A STUDY OF A SMALL, GASEOUS-FUELED ROTARY ENGINE

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A STUDY OF A SMALL, GASEOUS-FUELED ROTARY ENGINE

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I wish to express my sincere appreciation to my mother and father for the sincere interest they continually showed.

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SUMMARY

This work is a study of an air-cooled, gaseous powered Wankel rotary engine and predicts the feasibility and reliability of using an engine of this type as a stationary prime mover, such as might be found in a gaseous powered heat pump. A Sachs Model KM 37 Wankel engine was used for the study and was fueled with propane gas.

Power thermal efficiency, and exhaust emissions tests were made on the engine and air-fuel ratio, engine speed, and engine load were varied in an effort to determine the optimum operating conditions. It was found that at lean air-fuel ratios of approximately 18:1, maximum engine thermal efficiencies of 23 percent were developed without great sacrifice in engine performance. These compare with efficiencies of not greater than 17 percent produced by the engine operating on gasoline. Also, lean air-fuel ratios of about 18:1 produced lowest exhaust emissions under all conditions tested. The engine operating at 4 in. Hg manifold vacuum and 3500 RPM produced 20.9 gm HC/HP-hr, 6.4 gm CO/HP-hr, and .60 gm NO_x/HP-hr.

The reliability of the engine was determined by a seal wear study conducted on the rotor gas seals. The engine was equipped with special apparatus to allow the engine to be operated 24 hours a day, unattended. At predetermined intervals, the engine was disassembled and the accumulated seal wear was measured. These measurements provided curves depicting seal wear as a function of time. The extrapolation of these curves to the point where seal replacement is recommended by the engine

manufacturer provides an estimate of expected seal replacement intervals. These values are 33,000 hours for the apex seals, 19,000 hours for the corner seals, and 53,000 hours for the side seals.

Judging from all of the tests conducted, it appears that a rotary engine similar to the type tested shows great potential for being a very satisfactory stationary prime mover.

CHAPTER I

INTRODUCTION

The Energy Problem

As the population of the country increases and the standard of living continues to rise, the energy demand, too, increases at a phenomenal rate. Unfortunately, with this upward climbing demand for energy comes the decreasing availability of natural, usable energy resources of all types, and unless action is taken immediately, a serious energy shortage will inevitably develop in the near future. One of three things must be done to prevent this shortage. These are:

1. Cut down on the total consumer demand.

2. Develop plentiful, new resources, capable of supplying the entire population with energy for years to come.

3. Utilize the present fossil fuel supply in more efficient processes in order to extend the energy resources without cutting down on the demands.

To decrease the energy demand would undoubtably create a substantial decrease in the standard of living for all economic classes and would be a very unpopular solution. It can be shown that the standard of living of a country is directly proportional to the per capita energy consumption rates (1). The United States has the highest per capita rate and consequently the highest living standard.

New resources, such as nuclear and solar power, will inevitably have to be developed, but much more time and technology is needed to make them a real and lasting solution. Nuclear power is of course already existant in some power stations in the country, but due to its poor thermal efficiency creates a severe thermal pollution problem. It is also economically impossible to build enough nuclear power plants to take over a significant percentage of the energy demand before the year 2000.

Solar power would be ideal as it would completely eliminate the thermal pollution problem and has a fuel cost of zero. At present, however, solar energy is extremely costly to collect and until more inexpensive and efficient collection means are made available, this source also cannot be considered as a solution to the energy shortage before the year 2000.

Obviously, the more efficient utilization of our fossil fuels is the solution which now needs the most vigorous investigation. The fossil fuels being used in this country are coal, oil, and gaseous fuels. Of these, gaseous fuels are in the biggest danger of depletion. The ratio of the gaseous fuel supply over the amount consumed is about 12 which is smaller than with any of the other fuels (2). This fact, along with the fact that these fuels are the cleanest available and easiest to handle indicate that a strong effort must be made to conserve them for as long as possible.

Gaseous Prime Movers

One method of conserving the gaseous fuel supply is the development of efficient, reliable gaseous powered I.C. engines to be used as prime movers for stationary application. There is much evidence to indicate that in many instances of on-site gas utilization by a prime mover the overall energy system efficiency is greater than an equivalently powered electrical prime mover.

One application that must be carefully considered is comfort heating with a heat pump, since over 30 percent of the gaseous fuel supply is used for heating purposes. The heat pump concept of pumping heat from cool room to the warm outside air in the summer, and from the cool outside air into the warm inside room in the winter is well known. In either mode of operation, the system utilizes the conventional compressor driven air-conditioning system. The evaporator and condensor merely exchange their functions with the use of valves.

Heat pumps are extremely desirable for heating since they pump more energy inside the heated space than is required to drive the system. Therefore, when comparing electrically driven heat pump heating to electrical resistance heating, less electricity is required by the electric heat pump to maintain a constant, warm temperature inside.

By far, the majority of heat pumps found in the country today are electrically powered. However, there are two primary disadvantages found when using the electric heat pump. The first is that the relationship between the heating capacity and cooling capacity is given by the equation (3):

$$\frac{\text{Heating Capacity}}{\text{Cooling Capacity}} = \frac{1}{COP_{c/w}} + 1$$

where $COP_{c/w}$ is the cooling coefficient of performance of the pump and is defined by

$$COP_{c/w} = \frac{\text{Cooling Capacity}}{\text{Work Required}}$$

The $COP_{c/w}$ value is typically around three. Therefore, the heating capacity is only about 33 percent larger than the cooling capacity. Since, depending on location, heating demands are around 50 percent to 1000 percent higher than cooling demands (3), an electric heat pump must be sized so that it either a) has a much larger cooling capacity than is necessary, resulting in a much greater initial expense or, b) has a heat output considerably less than the heating demand. If the latter choice is made, a second heating system to supplement the heat pump system must be added. Neither of the choices is therefore desirable.

The second disadvantage comes from the fact that during the heating mode, the temperature of the evaporator coil must be below the atmospheric temperature in order for heat transfer to take place in the proper direction. Under certain conditions, the coil temperature is below both the dewpoint temperature and the freezing temperature of water and ice forms on the coil, blocking the flow of air. In this case, resistance heating must be supplied to the coil to melt the ice, thus reducing the advantage of the heat pump over ordinary resistance heating.

The gaseous powered I.C. engine driven heat pump can overcome both of these disadvantages as well as furnish high efficiency air-conditioning. For a gaseous powered heat pump operating in the heating mode, the following equation may be used (3):

$$COP_{h/g} = \frac{Q_{heating}}{Q_{gas}} = \eta_{eng}(1 + COP_{c/w}) + (1 - \eta_{eng})F_{rec}$$

In this equation, $Q_{heating}$ represents the heat pumped into the heated space, Q_{gas} represents the total heat liberated by the burned gas, η_{eng} is the engine thermal efficiency which is approximately 20 to 25 percent, and F_{rec} is the fraction of the waste heat from the engine which is recovered and put back into the heated space. The utilization of the waste heat from a gaseous heat pump would help eliminate the disadvantages mentioned before in electric pumps. An electric motor has a very high efficiency and produces essentially no waste heat itself. However, the efficiency of the power plant that produces the electricity so efficiently consumed is only about 30 percent, thus 70 percent of the total energy consumed is dumped into the atmosphere or rivers without being utilized. The utilization of waste heat produced by a gaseous heat pump can help give the pump much more heating capacity than an electric pump of the same size. Also, some of the waste heat can be directed to the evaporator coil to help reduce the freezing problem mentioned before.

Figure 1 shows $COP_{h/g}$ as a function of the waste heat recovery factor, assuming engine thermal efficiencies of 20 and 25 percent and $COP_{c/w}$ values of two, three, and four. Also shown is the range of $COP_{h/g}$ values for gaseous space heaters. The values on the graph can be compared to an overall COP of about .6 for an electrically driven heat pump (4).

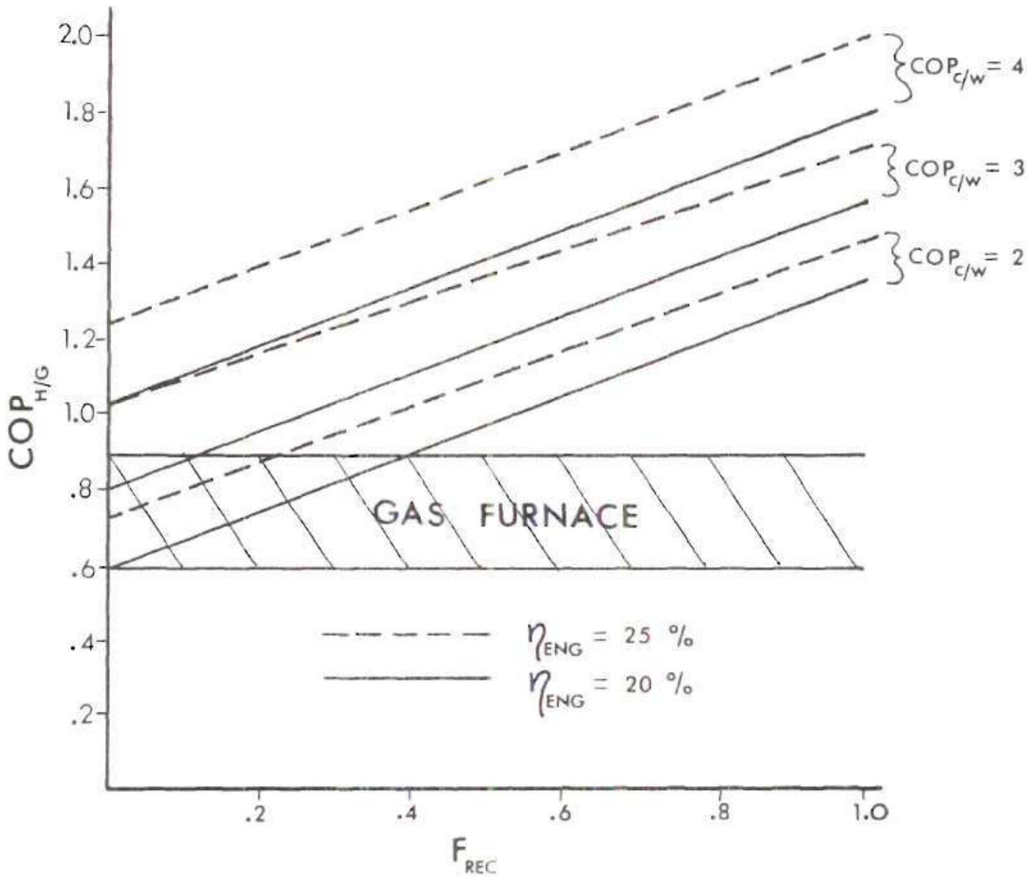

Figure 1. $COP_{h/g}$ Variation with Heat Recovery Factor

The $COP_{c/w}$ values are higher in areas of the country where the temperature differences between inside and outside are the smallest. In areas with mild winters, the $COP_{c/w}$ values may approach four. Where temperatures reach $0^{o}F$, the values decrease to possibly below two.

As can be seen, for a given quantity of gas as much as 90 percent more heat can be put into a heated space with a gaseous heat pump than with a conventional gas furnace.

Another advantage of the gaseous heat pump is the ability to control the engine speed. Increasing the engine speed increases the pressure drop in the system and consequently increases the temperature difference between the condensor and evaporator. If the temperature difference between inside and outside is large, the engine can be sped up and if the temperature difference is small, the engine can be slowed down, thus enabling the operation of the system at the maximum COP under a variety of temperature conditions.

Thus far, the discussion of gaseous powered I.C. engines as stationary prime movers has been limited to operation in heat pumps. There are, however, many other applications which could put this engine to good and efficient use.

One such application would be in a remote electric generating station. Many recreation homes are built away from main power lines and, depending on the location of such houses, the cost of electrical service can easily become prohibitive. In these cases, electricity can only be practically provided by a home generating system. An efficient, reliable, propane powered I.C. engine can be ideal to drive such a system. This same idea could also be used to provide low cost electricity for today's recreational vehicles which, in most cases, already have adequate propane tanks installed.

The success of using such an engine in any application is directly dependent upon the reliability of the engine. Depending on the particular use involved, the engine must be designed to operate prescribed minimum length of time before any maintenance whatsoever is required. Excess maintenance is extremely bothersome to the consumer but, more importantly, destroys the favorable economics of such a system.

The Wankel

One type of engine that shows promise for good reliability in stationary operation is the Wankel rotary engine. The Wankel posseses certain inherent attributes which indicate that it could have a much lower maintenance frequency and longer time between overhauls than an equivalently powered piston engine operating under the same conditions.

Figure 2 shows the basics of a single rotor Wankel. In this engine, there are fundamentally only two moving parts, the eccentric shaft and the rotor. No connecting rods are needed since the rotor revolves directly around the eccentric shaft and no valve operating mechanisms are needed since the engine is port timed. Power is transmitted directly to the output shaft by the rotor, which is kept properly positioned by the internal and external timing gears shown.

The Wankel operates on the Otto cycle and has the same processes as a conventional four stroke engine. Figure 3 depicts the sequential cycles of the rotary engine. As can be seen, since the rotor is three sided, there are three separate power impulses per rotor revolution. However, since the eccentric shaft rotates three times for each rotor revolution for a single rotor engine, there is only one power impulse per shaft revolution.

The simplicity of design of the Wankel as compared to conventional engines makes the future of the use of rotary engines in stationary operation look very bright. The fewer number of moving parts makes necessary maintenance tasks much less difficult and, in the larger engines, causes a substantial reduction of weight. In the 200 horsepower range, a rotary engine has only about 60 percent of the weight of a six

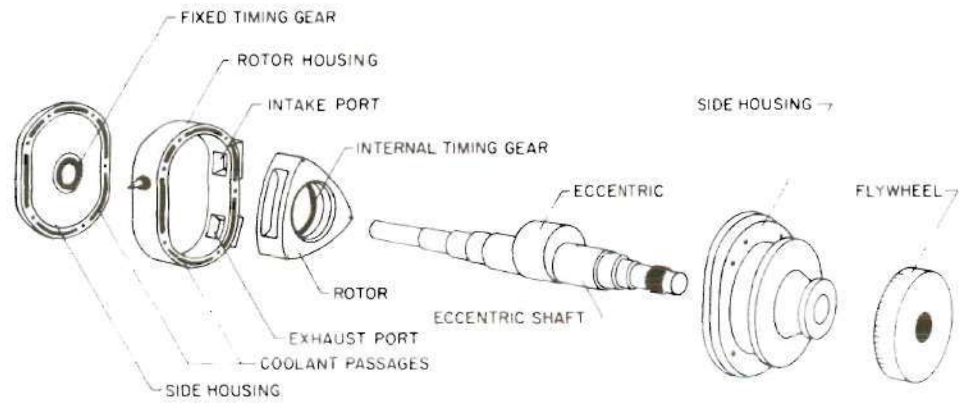

Figure 2. Basics of a Single Rotor Wankel Engine

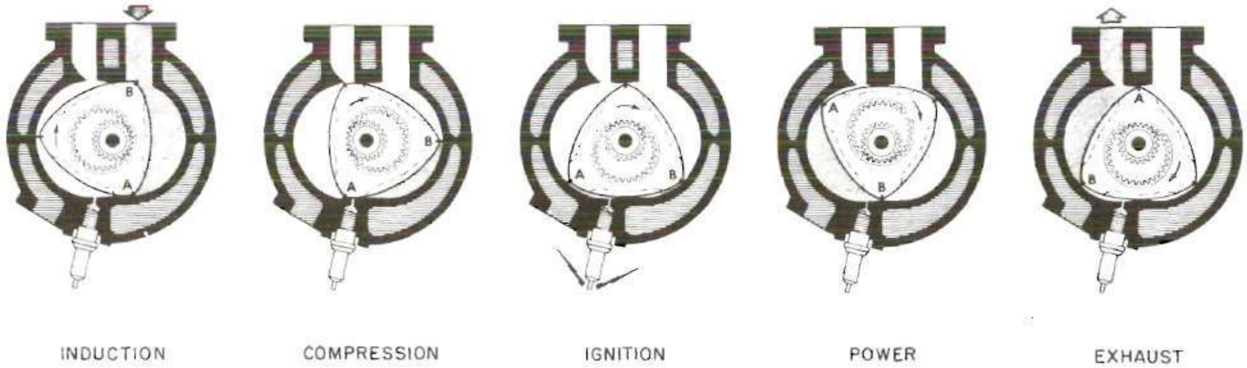

Figure 3. Wankel Engine Cycle

cylinder, four cycle piston engine of the same power. In the smaller engines of approximately ten horsepower, however, this advantage diminishes (5).

Another important advantage of the rotary engine over the piston engine is a much lower vibration level. A one cylinder, two stroke engine having a displacement of 12.2 cubic inches and a single rotor Wankel having a chamber volume of 6.6 cubic inches were compared to demonstrate this fact. It was shown that the amplitude of vibration of the piston engine was over five times that of the Wankel (5). Vibration is a serious problem and can cause increased fatigue failure rates both internal and external to the system. Bearing wear rates also increase rapidly with increased vibration.

A primary disadvantage of the rotary engine in the past has been the relatively short rotor seal life attained when operating on gasoline. The seals, not shown in Figure 2, are located on the rotor and prevent combustion gases from escaping from the combustion chamber. Reports show that the seals at the apexes of the rotor of a Sachs air-cooled Wankel decreased in radial dimension at a rate of approximately eight micro-inches per hour (5). With a maximum acceptable seal wear of about .020 inches, it can be seen that only about 2600 engine hours are possible before seal replacement is necessary. This constitutes a major maintenance task, somewhat similar to piston ring replacement in a conventional engine. Twenty-six hundred hours between overhauls is unsatisfactory for most applications of stationary prime movers.

From preliminary observations in Wankel engine tests in the School of Mechanical Engineering at Georgia Tech, a much longer period of time between overhauls appears possible from the Wankel when it is operated on gaseous fuels. The particular engine mentioned above, as is common with most small rotary engines, is normally lubricated with a 50:1 mixture of gasoline and oil. This results in an obvious dilution of the oil and possibly causes poorer lubricating properties. An engine of this type burning gaseous fuels would have to be lubricated by some type of oil injection method which would result in unaltered lubricating properties and, hopefully, less friction at the sealing interface and longer seal life. Also, gaseous fuels produce much less carbon in the engine than does gasoline. Again, less carbon means less abrasion and increased seal life.

The seal wear problem is essentially the only problem affecting the reliability of the Wankel engine in stationary application, as there are extremely low failure rates and long lifetimes associated with all the other parts. A good reliability test, therefore, would be a long term seal wear study. In such a study, a rotary engine would be run continuously for a long period of time, stopping it only for disassembly and measurement of seal dimensions.

Pollution Consideration

No power source can be considered exclusive of the effect it might have on the environment. A prime mover, no matter how efficient and reliable it may be, is not satisfactory if the ultimate exhaust emissions from it are above the acceptable environmental standards. Therefore, in the development of gaseous powered prime movers, the exhaust products must be monitored under as many operating conditions as possible to ensure that an air quality problem is not created.

Purpose of Research

This thesis is an effort to study a gaseous fueled, air-cooled Wankel engine and determine the feasibility of using such an engine as a stationary prime mover. Discussed and explained in detail are the set up and conduction of a long term seal wear study, power and efficiency tests, and extensive exhaust emissions tests. Data obtained is presented in both tabular and graphical form, and a conclusion indicating the feasibility and practicality of such an engine is discussed.

CHAPTER II

DESCRIPTION OF EQUIPMENT

The wear study, efficiency tests, and exhaust emissions tests conducted in this thesis required the installation and use of a varied array of laboratory apparatus. In the wear study especially, since an operator was not able to be present much of the time, several automatic shutoff and warning devices had to be designed and built to keep the engine from damaging itself during possible unscheduled changes in operating conditions. It is considered that all equipment used was essential in the collection of accurate data, and a detailed description of each item and its use will be given in the remainder of this chapter.

Basic Engine-Dynamometer Test Setup

The engine tested was a Sachs, single rotor, Wankel utility engine. It was basically aluminum in construction, with exception of the rotor, eccentric drive, and wear surfaces, and had a dry weight of approximately 34 pounds. More complete engine data are shown in Table 1 (6).

The engine carburetor came equipped with a spring loaded throttle adjustment lever which enabled course manual adjustment of power and speed. Since precise positioning and very small changes of throttle setting were required in the tests conducted, a precision throttle control was designed and installed. Rotation of a large knob then produced precision movement of the carburetor throttle plate.

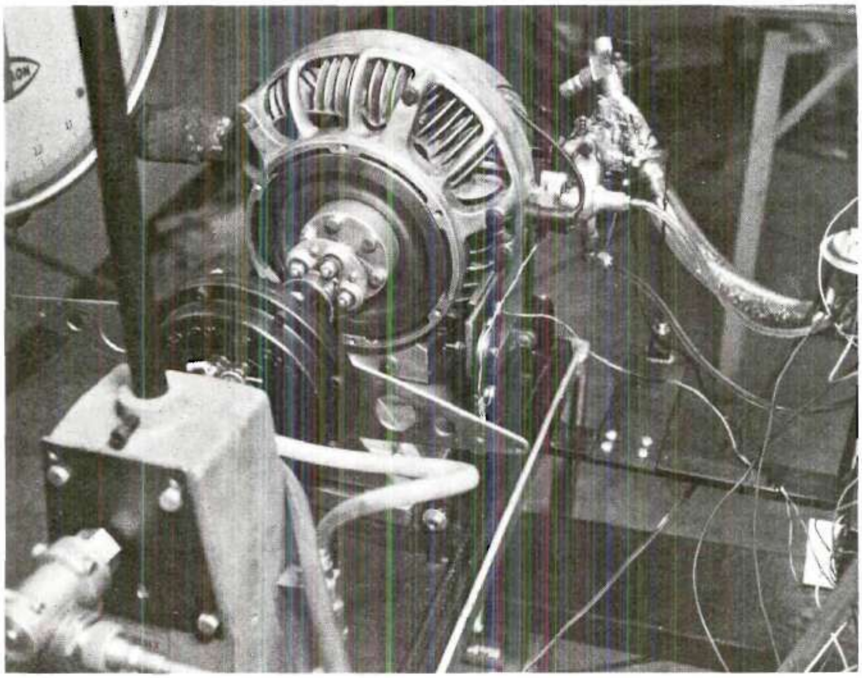

Figure 4. Basic Engine-Dynamometer Setup

Table 1. Test Engine Data

Type	Sachs Model KM 37 Wankel
Maximum RPM	5000
Output	5.5 horsepower at 4900 RPM
Cooling	Air cooling by fan
Compression ratio	8.5:1
Chamber volume	108 cc (6.59 cubic inches)
Maximum compression	175 psig at 1500 RPM
Ignition	Magneto
Firing position	10 -12 BTDC
Weight	34.2 pounds, dry

Originally, the test engine was started manually and came equipped only with a recoil type rope starter. It was discovered, however, that after altering air-fuel ratios or other parameters, at times the engine became almost impossible to start. An electric starter was therefore built and installed on the engine. An adapter and V-belt pulley were bolted to the cooling fan hub in a way not to inhibit the flow of air through the fan and a Ford 12 volt starter motor was fitted with another V-belt pulley. Pulley sizes were important and were designed to turn the engine at 1500 RPM. The starter motor was mounted below the fan pulley to a spring supported platform which was pivoted at a hinge bolted to the engine support frame. A starter relay was wired to the motor and activated by a spring loaded toggle switch. To start the engine, a V-belt was placed over the two pulleys and the starter motor platform was pushed down to tighten the belt. Pressing the toggle switch caused the starter motor to turn the engine and after the engine started, the toggle switch and platform were released and the belt was removed from the pulleys.

Initially, the engine exhaust was carried outside the lab through a low restriction, $1\frac{1}{2}$ in. flexible pipe welded to the outlet of the stock muffler which was bolted to the exhaust port. Later, the stock muffler was removed and the pipe was fastened directly to the port by means of a specially machined fitting. This was done to enable the construction of a sound absorbing housing completely around the engine. A new, small muffler was then welded to the exit end of the pipe to reduce the noise levels outside the building.

Power produced by the engine was absorbed by a Go-Power Model D-9 water dynamometer coupled directly to the output shaft of the engine. A Morse "Morflex" coupling was used to provide a firm, yet slightly flexible junction. Load was varied by changing the flow rate of water through the dynamometer. Precise flow rates were required to produce constant loads and these were obtained with the use of a two-stage water pressure regulation system. Water from the building supply passed through a Watts 25-80 psig regulator and a Fisher 5-15 psig regulator piped in series. It then passed through a Whity precision needle valve into the entrance of the dynamometer housing. The two-stage reduction eliminated inlet pressure changes due to variations in supply pressures and enabled a particular valve setting to dictate a constant water flow rate and load.

Torque values were determined by a 0-25 pound spring scale fastened between the arm of the dynamometer and a supporting bracket. The scale had a 12 in. face and was graduated in .05 pound increments which enabled precise determination of torque and horsepower (as in Appendix A). The arm length gave the dynamometer system a constant of 8266; i.e.,

$$\text{Horsepower} = \frac{\text{Scale Reading x RPM}}{8266}$$

The engine and dynamometer setup were firmly bolted to an aluminum plate designed for this system by Go-Power. This plate was supported about three feet from the floor by the engine support frame which was made of angle iron and was bolted directly to the floor.

A control and instrumentation board was attached to the support frame by means of two angle iron brackets. On this board was mounted the basic engine instrumentation which consisted basically of an AC 0-8000 RPM mechanical tachometer, a Heathkit 0-9000 RPM electronic tachometer, a 12 in. mercury U-tube manometer for measuring manifold vacuum, a Stewart-Warner engine hourmeter, and various other valves and switches for speed, power, and oiling rate variation and shutoff.

Vacuum Operated Micro-Switch Assembly

In conducting the seal wear study, the large number of engine hours necessary prohibited the constant attendance of an operator. For this reason, it was important to have several safety shutoff and warning devices incorporated in the setup at times of unscheduled changes in operating conditions or in case of engine failure. Some of these devices, all of which will be explained in detail later, were activated by a vacuum operated micro-switch assembly connected to the intake manifold.

The micro-switch used was the push button type and contained two normally open and one normally closed circuits. This switch was mounted by a special bracket to the moving arm of a spark advance diaphragm assembly obtained from a Chev rolet V-8 engine.

When the engine was running, the manifold vacuum was transmitted through tygon tubing to the diaphragm and caused the arm fastened to the diaphragm to be pulled away from the button of the micro-switch. In this position the normally open and normally closed circuits were open and closed respectively. When the engine stopped or manifold vacuum was destroyed for other reasons, a spring internal to the diaphragm assembly caused the diaphragm arm to depress the button on the switch and reverse the continuity of the circuits. This action and the number of circuits available on the switch gave great flexibility in the type and number of safety devices employed.

Engine Lubrication System

When using gasoline as fuel, this particular engine is lubricated by using a 50:1 mixture of gasoline and oil. When using gaseous fuel, however, this procedure must be abandoned in favor of some type of direct oil injection method. Figure 5 shows a schmatic of the gravity feed system used in all phases of the propane testing.

Shell SAE 30 Rotella oil was used for lubrication and was stored in a one gallon metal container which was supported three feet above the carburetor. This container was connected to a one pint Alemite machine oiler. The oiler was sealed off at the top to prevent leakage of oil due to the head pressure of the reservoir. Tygon tubing was used to connect in series the oiler, a solenoid valve, a needle valve, and a venturi through which intake air passed.

The oil flowed from the reservoir to the machine oiler where it fell, drop by drop, through the oiler sight glass, solenoid valve, needle valve, and into the venturi. Here it was taken into the engine

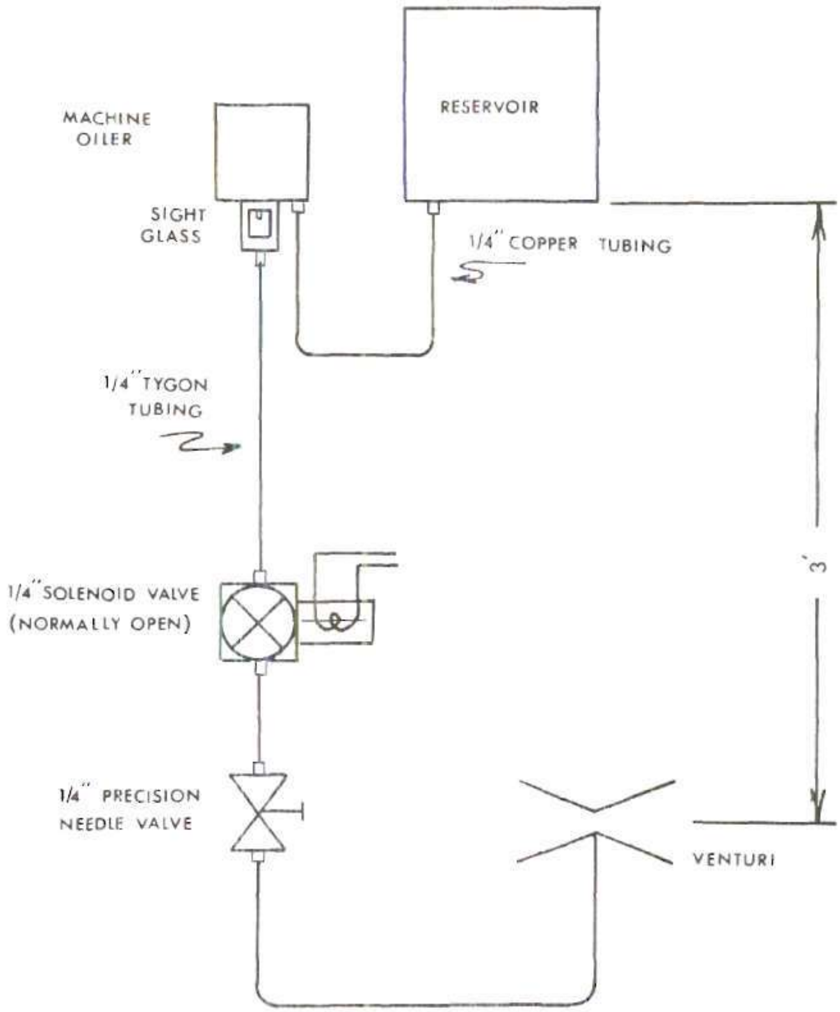

Figure 5. Engine Oiling System

with the air-fuel mixture.

The 12-volt normally open oiling solenoid valve was activated by a 12-volt relay wired through the vacuum operated micro-switch previously described. A three-amp battery charger was used as the power source for both the valve and the relay. The purpose of the solenoid valve was to stop the flow of oil into the engine during an unscheduled shutdown with no operator present. A toggle switch was placed in the relay circuit to override the solenoid valve when necessary. The wiring diagram for this circuit and other circuits is shown in Figure 6.

Regulation of the oiling rate was provided by the Whitey needle valve located downstream of the solenoid valve. Flow rates of zero up to 2 drops/sec were accurately obtained with this valve and during scheduled shutdown it was used to shut off the oil supply entirely.

Fuel System

Propane was used exclusively for fuel, with the exception of an initial power and efficiency test conducted with gasoline. Figure 7 shows a schematic of the fuel system used for the seal wear test. Minor modifications to this system were made for obtaining the exhaust emissions and efficiency data.

Propane was stored in a 100 gallon steel tank located directly outside the building. Liquid propane at 130 psig passed from the tank into an Impco Model E pressure regulator which gassefied the liquid and dropped the pressure to -1/4 in. W.G. Tap water was also passed through the regulator to supply heat to vaporize the liquid. From the regulator, the gas flowed through 1/4 in. fuel line to the flow measuring apparatus

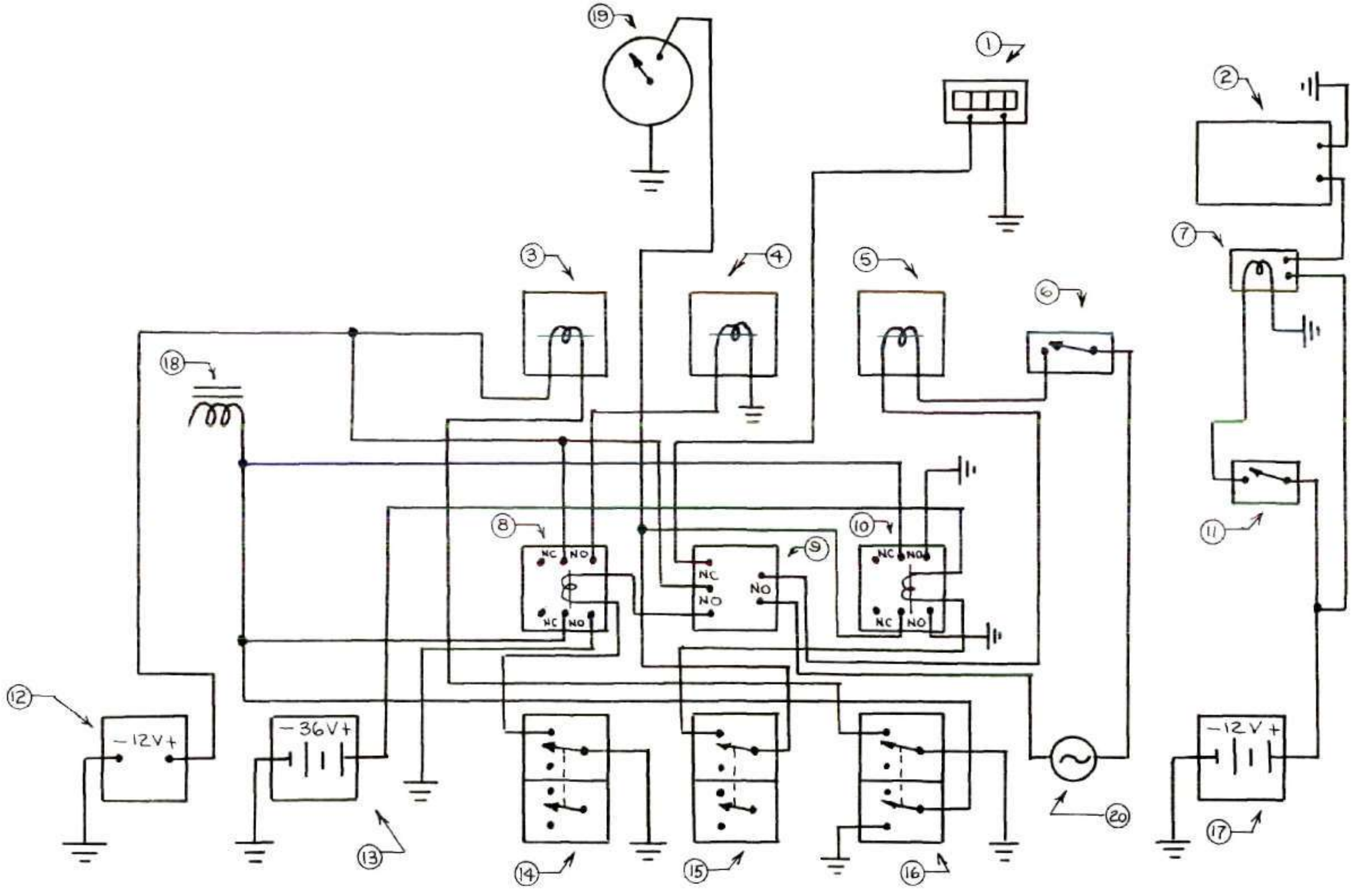

Figure 6. Complete System Wiring Diagram

Legend for Figure 6:

1. Engine Hourmeter
2. Starter Motor
3. Oiling Solenoid Valve
4. Propane Solenoid Valve (fuel line)
5. Telephone Signaling Device Solenoid
6. Telephone Signaling Device De-energizing Switch
7. Starter Relay
8. DPDT 12-Volt Relay
9. Vacuum Operated Micro-Switch
10. DPDT 36 Volt Relay
11. Starter Switch
12. 12-Volt Battery Charger
13. 36-Volt Battery Source
14. DPDT Toggle Switch (oil shutoff system)
15. DPDT Toggle Switch (high RPM shutoff system)
16. DPDT Toggle Switch (ignition)
17. 12 Volt Auto Battery
18. Primary Coil of Engine Magneto
19. Mechanical Tachometer (wired for high RPM shutoff)
20. 110 Volt AC Source

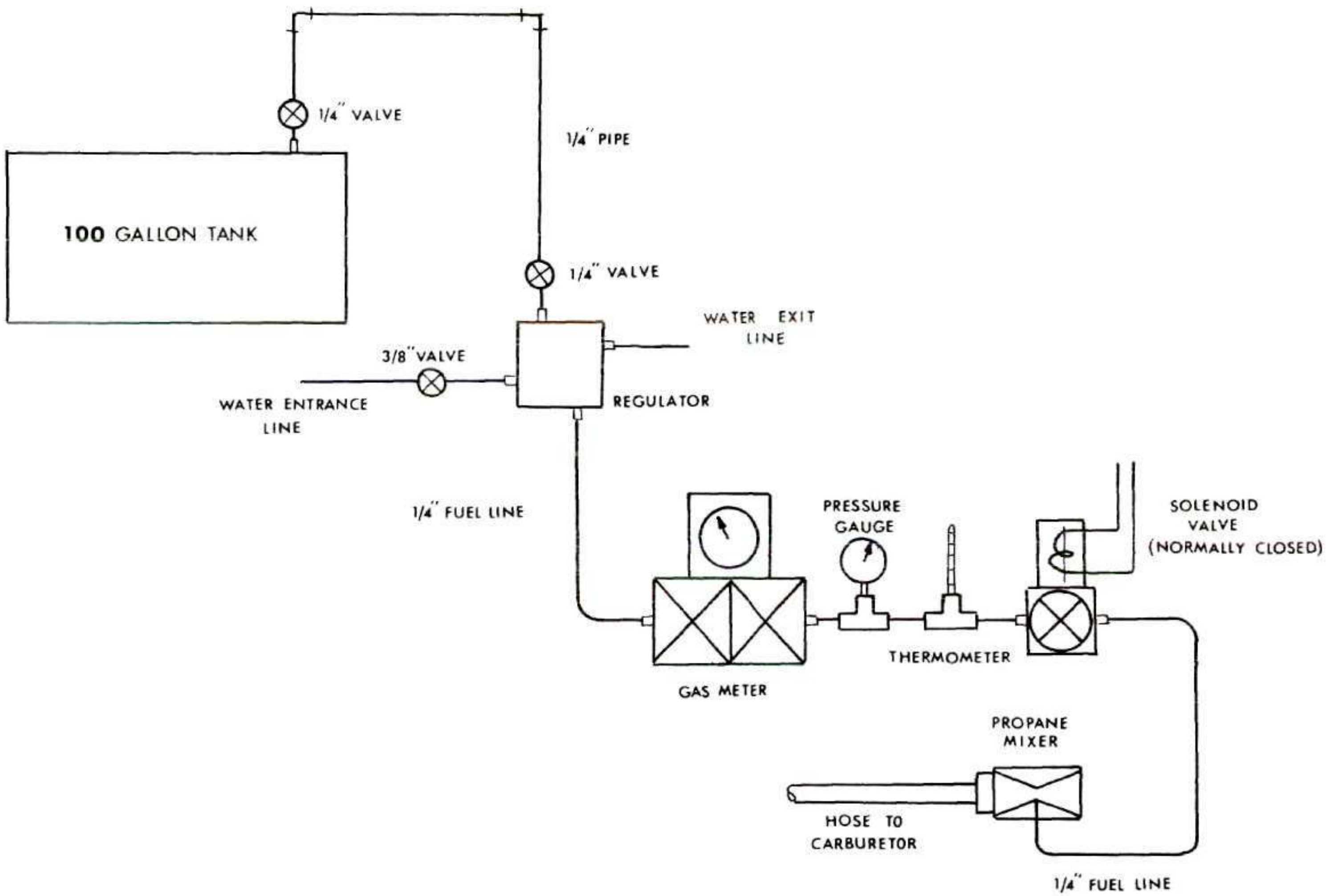

Figure 7. Gaseous Fuel System

and then through a 12-volt normally closed solenoid valve. The solenoid valve was wired up through a circuit of the ignition switch and was used to prevent fuel flow when the engine was not running. From the solenoid valve, the fuel flowed into the air-fuel mixer. The air-fuel mixture then passed through a plastic hose to the inlet of the carburetor.

The flow measuring apparatus consisted of a gas meter, pressure gauge, and thermometer, all located in series. A Sprague Model 175 gas meter was used and was capable of measuring volumes to 1/100 cubic foot. The pressure gauge had a full scale reading of 5 psig and the thermometer registered temperature in degrees F. Knowing the volume flow rate, temperature, and pressure, the fuel mass flow rate could easily be calculated (Appendix C).

The gas mixer was a Beam Model 1180 propane mixer. Passage of intake air through a venturi in this mixer caused a low pressure area to be formed and enabled fuel to flow from the regulator to the mixer and into the engine. Air-fuel ratios could be crudely adjusted by turning a coarse needle valve located on the fuel inlet of the mixer.

When conducting exhaust emissions and efficiency tests, a more precise method of adjusting and setting air-fuel ratios was required. This was accomplished by slightly modifying the Impco pressure regulator and by inserting a precision Whitey needle valve just upstream of the mixer. A purge button was located on the regulator and when depressed, created a higher, constant fuel pressure of up to 5 psig. A control screw to depress the button was mounted in a bracket to the face of the regulator and enabled the manual selection and setting of any desired pressure. This higher pressure used in conjunction with the throttling capabilities of the needle valve made possible accurate attainment and easy relocation of air-fuel ratios.

High RPM Shutoff

One of the important safety devices installed was an electrical high speed shutoff which was used to stop the engine if the RPM exceeded a certain predetermined value which migh occur due to unscheduled load reduction or increased throttle opening. Use of the device was limited primarily to times during the seal wear study when an operator was not present.

The AC mechanical tachometer was mounted in the instrument panel and was driven from the shaft of the dynamometer by a flexible cable and 2:1 speed reducer. The tachometer was calibrated to read actual engine RPM. A small hole was drilled in the glass cover plate of the tachometer and a wire was inserted through the hole in a manner such that the RPM indicating needle would strike it when moving up scale. Care was taken to ensure that the wire was completely insulated from the tachometer casing. The cover glass could be rotated to put the wire at any RPM that shutoff was desired. When the metal indicating needle touched the wire, it completed a circuit from the wire to the tachometer casing. This circuit allowed the energizing of a 36 volt double pole-double throw mineature relay. The primary coil of the ignition was wired to ground through one set of relay contacts and a relay hold-in circuit was wired through another. A toggle switch was placed in the solenoid circuit for both de-energizing and resetting the device. The wiring for this device can be seen in Figure 6.

When the engine RPM attained the preset value for shutoff, the indicating needle made contact with the wire and activated the solenoid in the relay, causing the primary ignition coil to short out and stop the engine. The hold-in circuit was necessary to prevent an on-off-on-off oscillation. After the engine stopped, the device could be either overridden or reset by throwing the toggle switch to the off position.

Remote Telephone Signaling System

As an attendant was not always present during hours of operation in the wear test, it was important to have some method of remotely determining whether or not the engine was running. A simple telephone signaling system was installed for this purpose. The schematic is shown in Figure 8.

A standard wall type telephone on the regular Bell Telephone circuit was used for the system. On the wall, directly under the telephone, was attached a special bracket on which a 110 volt solenoid and toggle switch were mounted and linked together. As can be seen from the schematic, the receiver hook was held down by a pin when the solenoid was not activated and was released when the pin was pulled upon activation. The toggle switch and solenoid were wired in series to a normally open contact of the vacuum operated micro-switch previously mentioned (see Figure 6). The toggle switch was used only to de-energize the circuit after activation to prevent overheating.

When the engine was running, the contact on the micro-switch was open and the receiver hook remained in the down position. The telephone number could then be dialed from any phone and a ringing signal would in-

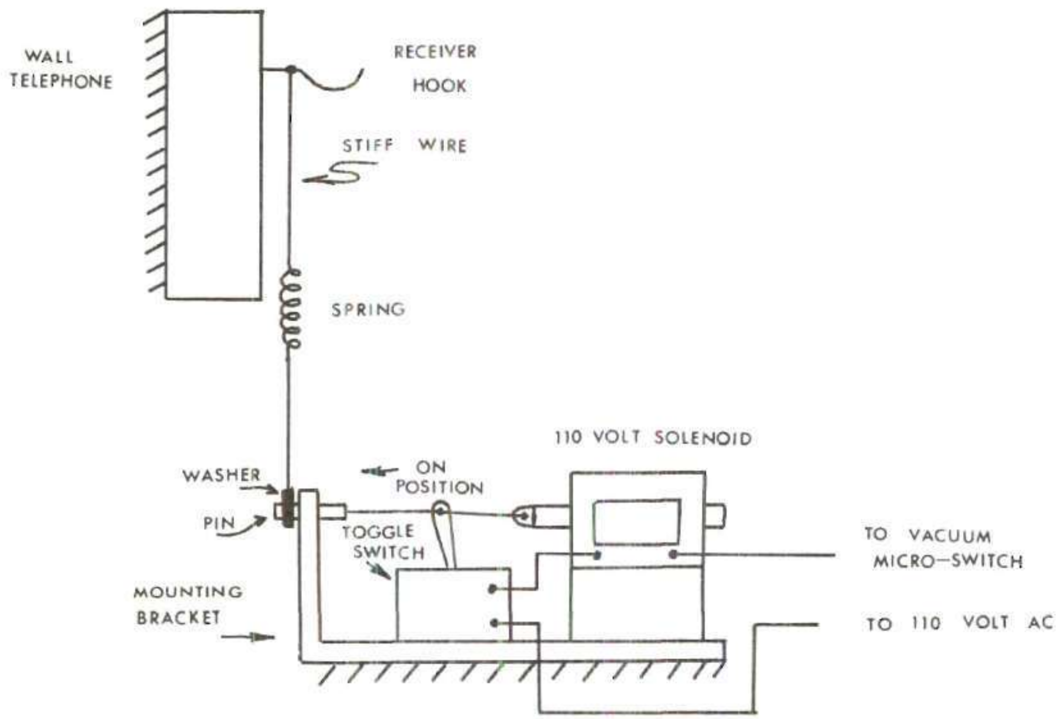

Figure 8. Remote Telephone Signaling System

Figure 9. Photograph of Remote Telephone Signaling System

dicate that the engine was still operating. Upon engine shutdown, the micro-switch contact would close, activating the solenoid and pulling the pin, thus releasing the receiver hook. If the number was then dialed, a busy signal would be heard, indicating the engine had stopped.

Speed Control

Even through the load on the engine could be held constant and the throttle control could be precisely set, the engine speed would, nevertheless, sometimes fluctuate. This was a serious problem during the wear test and to eliminate it, a speed control was designed and installed.

The engine was originally equipped with a top speed limiting device which consisted merely of a spring loaded air vane coupled through linkages to the throttle of the carburetor. When the engine was running, cooling air flowed past the vane and when a preset maximum RPM was reached, the pressure of the moving air on the vane overpowered the spring and caused the linkages to move and reduce the throttle setting, thus holding a constant top speed.

This top speed limiting device was modified to enable the setting and maintaining of any speed required. To do this, the spring force on the vane was merely partially counteracted by the force of gravity acting on a small cup filled with BB shot connected by a flexible wire to the arm of the air vane. Since the weight of the BB shot effectively reduced the spring force acting on the vane, it also lowered the maximum engine speed attainable. By opening the throttle control to its wide open position (WOT), the speed control was allowed to take over and by varying the amount of BB shot in the cup, any engine speed desired could be satisfactorily maintained. It should be noted, however, that this device was not a precision speed governor and worked properly only with a fairly constant engine load.

Engine Hourmeter

For the wear study, it was manditory to keep an exact record of the number of hours the engine was in operation. A Stewart-Warner 12-volt hourmeter was installed in the instrument panel for this purpose. The meter was wired in series with the 12-volt power source and a normally closed contact on the vacuum operated micro-switch (see Figure 6). This enabled the hourmeter only to operate only during the hours the engine was running.

Emissions Sampling and Measurement Apparatus

The laboratory was fully equipped with proper instrumentation to measure the exhaust emissions from the engine. All equipment was neatly installed in a special case called a sample train which was built earlier to facilitate the conduction of such tests. A flame ionization detector (FID) for measuring total hydrocarbons and non-dispersive infared (NDIR) analyzers for measuring carbon monoxide, carbon dioxide, and nitric oxide concentrations were the instruments integrated into this train.

Hydrocarbons (FID)

A Beckman Model 400 hydrocarbon analyzer was used to automatically and continuously measure the volume concentration of hydrocarbons (HC) in the engine exhaust gases by using the flame ionization method of detection. This device was later modified to also determine air-fuel ratios directly from intake gas mixture analysis.

The theory of operation of the FID unit is relatively simple. The sample in question passes directly into a hydrogen flame and is ionized. An ionization current is produced and is electrically monitored. Because this current varies linearly with the carbon concentration in the sample, with proper instrument calibration, HC concentrations can be read directly from a meter contained in the FID unit. Accuracies of $\pm$ 2% are claimed by the instrument manufacturer.

The fuel used for the hydrogen flame was 40 percent H_2 and 60 percent N_2 supplied at 30 psig. Air was also supplied at 20 psig. Internal pressure regulators to measure the fuel, air, and sample pressures were installed in the instrument. Fuel, air, and calibration (span) gases were stored in high pressure cylinders equipped with high pressure regulators.

Calibration of the analyzer was very simple. Compressed air was used to set the zero point and a span gas, consisting of dry nitrogen containing a known concentration of propane (C_3H_8) was used to set an upscale point. All span concentrations were multiplied by three to give instrument calibration on a per carbon basis. In the emissions data, however, these values were divided by six to give the hexane (C_6H_{14}) equivalent. The span gas used had a concentration of 9500 ppm propane (28500 ppm C).

As was stated, the FID unit was modified to measure air-fuel ratios directly from intake mixture analysis. The air-fuel ratio is a very important parameter and, in the past, has usually been determined either by direct measurement of air and fuel masses or by Spindt's

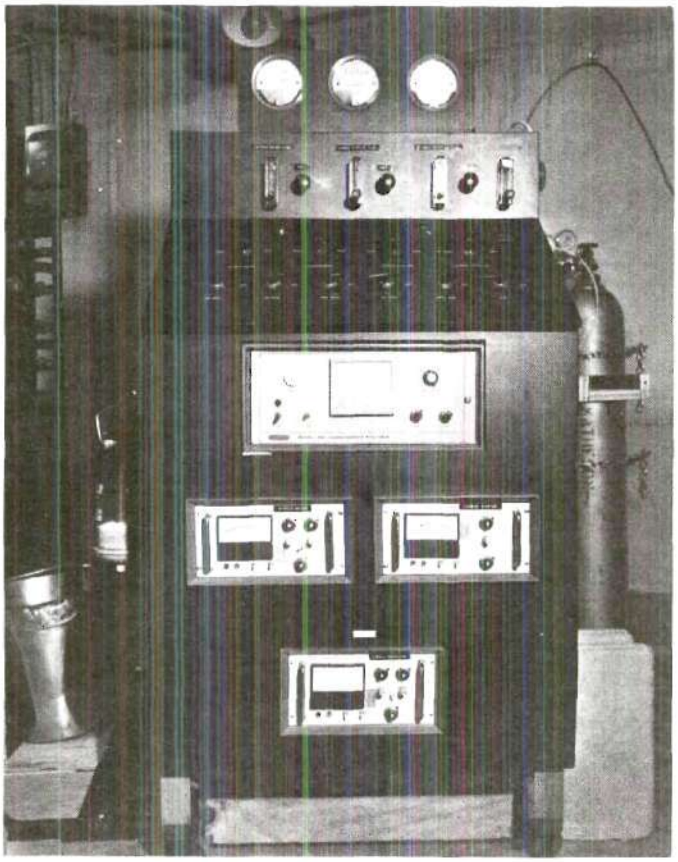

Figure 10. Sample Train

method of air-fuel ratio determination from exhaust gas analysis (7). These methods are extremely troublesome, however, and are often very inaccurate.

When using gaseous fuels, the air-fuel mixture is simply the mixture of a known hydrocarbon in air. If ideal gases are assumed, knowledge of the HC volume concentration will determine exactly what the air-fuel ratios are.

The modification of the FID analyzer consisted merely of extending the maximum range capabilities of the instrument so that it could be used to measure volumetric propane concentrations in the intake mixture. The stock analyzer comes with a maximum range of about 10 percent C. When using propane as fuel, however, HC concentrations in the air-fuel mixture often exceed 15 percent C. A minimum range increase of 50 percent is obviously necessary if this method of air-fuel ratio determination is to be satisfactory.

The sample gas is accurately metered into the hydrogen flame by a very simple method. The gas is forced by a pump on the sample train through a tube located in the analyzer. The pressure in the tube is held constant by an upstream pressure regulator. A capillary tube connects the larger tube with the hydrogen burner. The small diameter of the capillary tube causes an extremely small flow rate into the burner and consequently, most of the sample bypasses the hydrogen flame, unused.

A maximum range of 10 percent C is available in the stock instrument because concentrations above this value produce an ionization current in the flame not linearly proportional to the concentrations. If less sample flow to the burner can be achieved, however, higher concentrations can be passed through the analyzer without increase in ionization current.

Lengthening the capillary tube was one easy method of accurately cutting down the flow rate of sample into the flame. A capillary tube roughly twice the length of the original was installed and HC concentrations upwards of 20 percent were easily obtained with the unit without loss of linearity. Low level concentrations in the exhaust gases, however, could still be monitored by the use of a range selector switch located in the amplification circuit of the analyzer.

When the instrument was properly calibrated, the meter could be rescaled to read air-fuel ratios directly. The formula used for this is based on the ideal gas equation and can be found in Appendix E.

It was important that the air-fuel mixture be well mixed before entry into the analyzer as irroneous results would have been achieved otherwise. Good mixing was accomplished by the insertion of a mixing chamber between the propane mixer and the caburetor inlet. The chamber was made from an 18 in. length of clear plastic 1 in. hose. Turbulence in this hose was created by the insertion of a small piece of copper wool in the hose. A port through which the intake gas sample was taken was made in the hose just downstream of the copper wool.

Although very little sample actually passed into the hydrogen flame, a considerable amount of sample flowed through the bypass of the analyzer. When the air-fuel mixture sample was taken, the reduction of air-fuel mixture entering the engine was enough to cause a slight decrease in engine power output. This problem was eliminated by return-ing the unaltered bypass mixture to the mixing tube, downstream of where the sample was taken. A Whitey three-way valve was inserted in the return line to prevent the entrance of exhaust gas bypass into the carburetor during exhaust gas analysis.

NDIR Analyzers

Nitric oxide (NO), Carbon Dioxide (CO_2), and Carbon Monoxide (CO) concentrations in the exhaust gases were determined by Olson-Horiba Model AlA-2 NDIR analyzers. The principle of operation in all of these analyzers is the same. Infared light from two identical sources passes through a rotating, slotted disc where it is chopped, and then passes through two cells. The sample being analyzed passes through one cell and a reference gas remains in the other. Some of the infared radiation at a certain wavelength is absorbed in the sample cell and this is determined by a detector located at the end of the cells. The important thing is that the amount of radiation absorbed increases with the volume concentration of the particular pollutant under consideration. The detector causes an upscale reading of a meter in the instrument and this reading is applied to a special calibration curve to determine a value for the volume concentration. Accuracies of $\pm$ 1% are claimed by the manufacturer.

Calibration and operation of all the NDIR instruments was very similar. The only significant differences in the units were the number of concentration ranges available and the length and number of sample cells. Good documentation of instrument specifications and of step-by-step calibration and emissions measurement procedures for all of the analyzers can be found in references (8) and (9) and will not be discussed in detail in this text.

Sample Train System

As was stated before, all of the gas analyzing equipment was incorporated into one system called the sample train. The instruments were housed in a cabinet on which were mounted all necessary valves, rotameters, pressure gauges, and interconnecting tubing. Figure 11 shows the schematic of the entire system.

Samples tested were pulled through an ice bath before being pumped to the analyzers. The pump used was a Model MB 110-10 welded bellows vacuum pump/compressor which was manufactured by the Metal Bellows Corporation and was capable of supplying a maximum of 66 SCFH of gas. Selection of either exhaust or intake sample was made by proper manipulation of the two valves just upstream of the ice bath. The ice bath was made of a coil of 3/8 in. stainless steel tubing placed in a large stainless steel bucket. Ice was placed in the bucket to cool the exhaust gas sample and condense out most (essentially 100 percent) of the water vapor. A trap was placed directly downstream of the bath to collect the condensed water. The sample gas passed through the pump after going through the ice bath and then was directed to each of the different analyzers.

The HC portion of the sample passed through one gate of a Whitey five-way valve and through a Wilkerson Model 1049 particulate filter before entering into the FID unit. The other gates of the five-way valve were used to allow entry of the span and zero gases. Bypass from the analyzer was routed through a Whitey three-way valve either to the

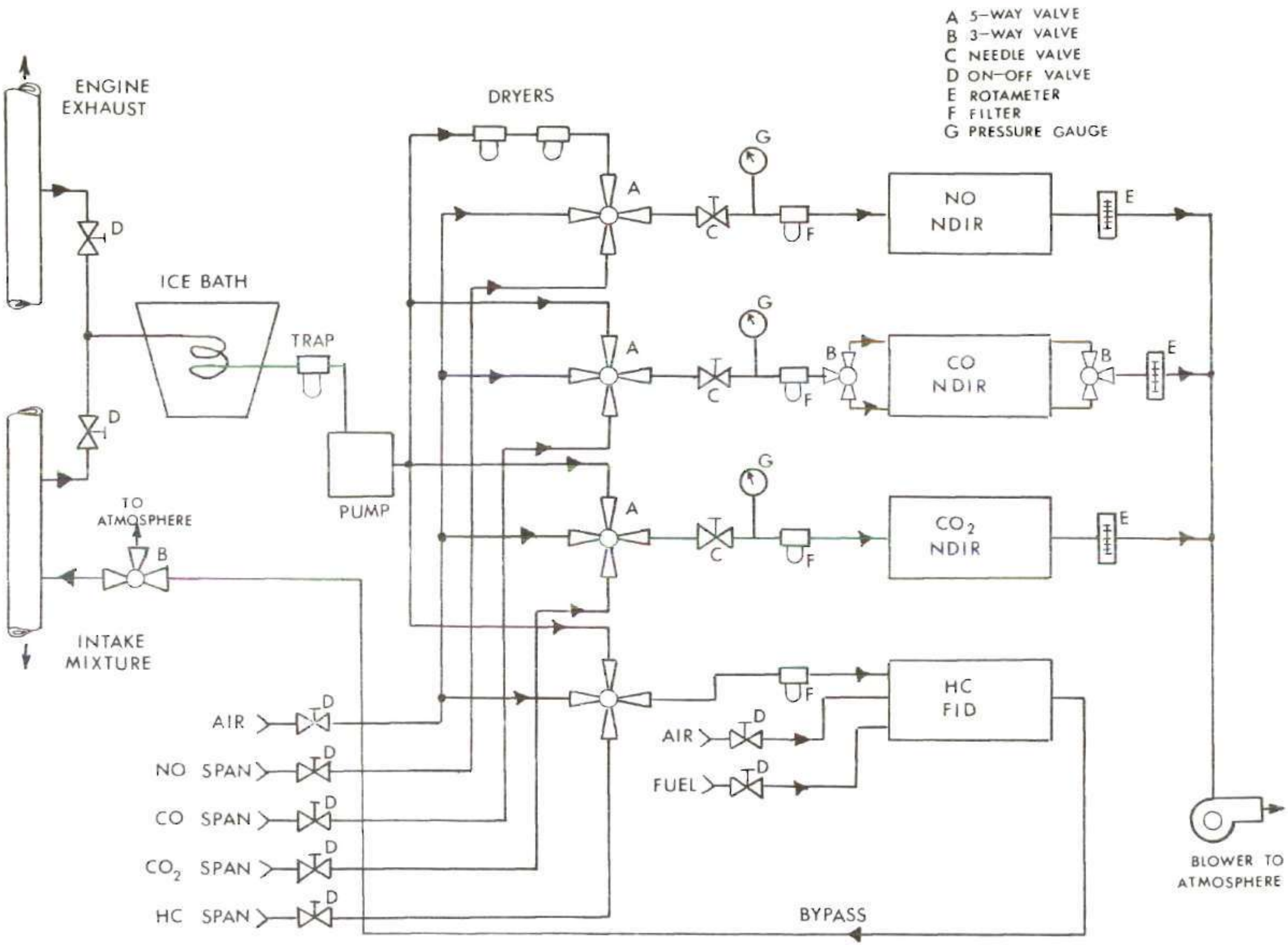

Figure 11. Sample Train Plumbing

atmosphere or back to the mixing chamber, depending on whether exhaust or intake samples were being considered.

The fraction of the exhaust gas sample which was pumped to the NO analyzer passed first through two Wilkerson Model 4001-2 dryers containing indicating silica gel. These were to remove any moisture not condensed out by the ice bath and were needed only for the NO unit. After the dryers, the sample passed through one gate of a Whitey five-way valve, a Wilkerson Model 1049 particulate filter, and then into the analyzer. Inlet sample pressure was measured by a Magnehelic 0-10 in. W.G. manometer. Flow rate was measured by a Brooks 2-18 SCFH rotameter just before the sample was exhausted into the atmosphere.

Plumbing of the CO and CO_2 units was almost identical to the NO analyzer. The only differences were that the silica gel dryers were not present, and for the CO analyzer, three-way valves had to be placed directly before and after the analyzer unit to allow for selection of one of the two available sample cells. The settings on these valves, however, never had to be changed during any of the tests conducted in this study.

All zero and span gases had to pass first through a Whitey off-on ball valve before entry into the five-way valves. All valves were made of stainless steel with the exception of the brass three-way valve on the bypass of the FID unit. Tubing was made of stainless steel, tygon, or nylon and Swagelock tube fittings were used exclusively.

CHAPTER III

TEST PROCEDURES

As the engine was not new when received and no information concerning previous engine operation was available, it was necessary to rebuild the engine in ensure valid emissions and wear data. Rebuilding consisted mainly of installing new apex and side seals and springs. Original corner seals had to be used as new ones could not be purchased, but they were, however, not badly worn. Other wear surfaces were inspected and were found to be in relatively good condition. New eccentric shaft bearings and a new rotor bearing were installed even though the original bearings showed no wear.

Power and Efficiency Test (Gasoline)

First to be conducted were power and efficiency tests using gasoline as fuel. These tests were not of prime importance, but were made merely to compare basic engine operation on gasoline with that on propane. In these tests, the stock carburetor was used and no attempt was made to determine the air-fuel ratios or hold them constant.

A 50 ml pipette and a stopwatch were used to measure fuel consumption rates. The pipette and the original engine gasoline tank were connected to the carburetor inlet by means of a three-way stopcock and tygon tubing. Connections were made as to enable the fuel to be drawn from either the tank or the pipette, simply by switching positions of the three-way stopcock. The stopcock was positioned to allow fuel to be drawn from the tank and the pipette was filled with gasoline. The engine was started and adjusted and when the desired operating conditions were attained, the stopcock was switched and the fuel was drawn from the pipette. Fuel consumption rates were determined by measuring the time for 50 ml of gasoline to be burned.

Tests were made at two different manifold vacuum settings and at four different speeds, thus 16 operating points were set. Manifold vacuums were read from the mercury manometer connected to the intake manifold. The operating points were set by adjusting the precision throttle control and dynamometer needle valve until proper loads and engine speeds were reached. Horsepower was calculated from the dynamometer scale and tachometer readings. From the fuel consumption and horsepower values, thermal efficiencies were easily calculated. Appendix B contains a sample calculation.

Measurement of Oiling Rate

As was described before, while operating on propane, the engine was lubricated by direct oil injection into a venturi. Since this was the only method of lubrication used, it was essential that ample oil be supplied to keep down excess wear and engine seizure.

The oiling rates used during propane operation were based on the oiling rates used with gasoline. The gasoline oiling rates were obtained for a variety of operating conditions simply by dividing the fuel consumption rates by 50. This gave oil consumption rates in ml/sec. Since the effectiveness of the injection method used was, at this time, unknown, a safety factor of 1.5 was applied to determine the propane oiling rates. This is to say that initially, 50 percent more oil was used during propane operation than during gasoline operation. This safety factor, however, was eliminated after the 415 hour tear down in the wear test.

Since the oil injection system visibly dispensed oil drop by drop, the flow rates in ml/sec were converted to a sec/drop basis. To do this, oil was allowed to drop at a constant rate of 1.18 sec/drop into a graduated cylinder. A time of 1725 seconds was required for 56 ml of oil to be dispensed. From these values, the drop volume was easily calculated to be .038 ml and oil drop rates for propane operation were calculated by the equation

$$\text{Drop Rate } \frac{\text{sec}}{\text{drop}} = \frac{.038}{\text{CRxSF}}$$

where CR is the oil consumption rate with gasoline in ml/sec and SF is the safety factor mentioned above.

Engine Exhaust Emissions Measurement

Engine exhaust emissions measurements were a very important part of this study and utmost care was taken during the experimentation to ensure accurate results. Effort was made to obtain data at as many operating conditions as possible in order to give a broad spectrum of emissions characteristics.

The parameters varied in the tests were air-fuel ratio, load setting, and engine speed. Loads set were no-load, 4 in. Hg manifold vacuum and WOT, engine speeds were 2000 RPM, 3500 RPM, and 5000 RPM, and air-fuel ratios were 13 (rich), 15.6 (stoichiometric), and 18 (lean).

A complete matrix of operating points was set and for each operating point the HC, CO, NO, and CO_2 concentrations in the exhaust were determined. Consequently, 3x3x3x4 or 108 data points were obtained for all of the emissions tests combined.

The first thing done before any data was recorded was the initial warm up and calibration of the exhaust analyzing equipment. While this was being done, the engine was started and allowed to warm up under idle conditions.

When everything was ready for testing, the valves on the lines from the exhaust pipe and fuel mixing chamber were closed and opened respectively to allow intake mixture to flow to the FID unit for air-fuel ratio determination. The flow was shut off to all of the other analyzers and the three-way valve on the bypass line was positioned to divert the bypass back to the engine. Air-fuel ratios were adjusted by the needle valve on the fuel line and were read directly on the meter in the HC analyzer. Load settings and engine speed settings were obtained by adjusting the dynamometer needle valve and precision throttle adjustment.

In the part load tests (4 in. Hg), considerable manipulation was often required to achieve the exact operating points desired. In the WOT tests, the throttle was adjusted to its maximum position and left there. This greatly simplified obtaining any operating point as it eliminated one variable. In the no-load tests, the dynamometer absorbing unit was removed from the engine and this, too, simplified operating point setting as the dynamometer needle valve required no adjustment. For no-load operation, since the dynamometer shaft was not present to drive the mechanical tachometer, an electronic tachometer was temporarily installed for engine speed determination.

After a desired operating point had been set and allowed to stabilize, the procedure was followed to take exhaust emissions readings. The valves on the lines from the mixing tube and exhaust pipe were closed and opened respectively to allow exhaust gases to be directed to the sample train. Appropriate valves on the sample train were adjusted to allow the gases to flow properly through all of the analyzers. The three-way valve on the FID bypass line was switched to allow the bypass exhaust gases to flow to the atmosphere instead of back to the carburetor.

When flow through the analyzers was proper, the meter readings on the CO, NO, CO_2, and HC units were recorded. After this, the necessary valves were changed to recheck the air-fuel ratio. If for some reason the air-fuel ratio had varied, the operating point was reset and the complete test was repeated.

When it was felt that satisfactory data had been obtained, the NDIR meter readings were converted by means of the instrument calibration curves to volume concentrations. Because the exhaust gases had to first pass through the ice bath before entry into the analyzers, essentially all water vapor in the exhaust was removed and thus concentrations obtained were on a dry basis. With exception of no-load data, these concentrations were corrected to a wet basis by the method used by Stivender (10) and with knowledge of the power output, fuel consumption, and air-fuel ratios, the emissions were converted to gm/HP-hr (Appendix D). Since NO is eventually converted to NO_2, all NO concentrations were converted to NO_2 concentrations by multiplying them by 46/30, the ratio of the molecular weights of NO_2 and NO.

Power and Efficiency Tests (Propane)

When conducting power and efficiency tests on propane, the starting and operating point setting procedure was identical to that of the propane exhaust emissions tests. Since no emissions tests were being conducted, however, the intake mixture was constantly monitored to determine air-fuel ratios. Fuel consumption rates were measured by recording the time for 1/2 cubic foot of propane to flow through the gas meter. Propane temperature and pressure were also recorded. Horsepower was calculated from the dynamometer scale readings and tachometer readings. From horsepower output, fuel volume consumption, temperature and pressure, thermal efficiencies were easily calculated (Appendix C).

Power and efficiency tests were only conducted on two load settings, 4 in. Hg. and WOT, and three speed settings, 2000 RPM, 3500 RPM, and 5000 RPM. No-load fuel consumption tests were not conducted due to the extremely small flow rates obtained with zero load and the inability to hold engine settings perfectly constant for very long periods of time.

Seal Wear Test

A wear test was conducted to determine the wearing rates on the rotor gas seals. As was stated before, excess seal wear has in the past been a problem of the Wankel engine while operating on gasoline. This test, therefore, was conducted entirely with propane to determine seal wear characteristics of a gaseous fueled engine.

On the KM 37, the rotor assembly is one of the more complicated parts of the engine. There are 15 different seals on the rotor: three apex seals, six side seals, and six corner seals. Under each apex and side seal there is one leaf type spring and under each corner seal there are two horseshoe type springs. Thus, including the rotor, there are 37 individual pieces.

The purpose of the apex seals is to provide leak free contact between the apexes of the rotor and the epitrochoidal bore. Wear on the apex seals is generally greater than any of the other seals due primarily to lubrication and temperature problems. Side seals are used to prevent the leakage of combustion gases between the rotor sides and side housings. Corner seals are used to prevent leakage past the ends of the apex seals where the side seals are unable to extend. The springs under all the seals are used to push the seals firmly against the sealing surfaces and to compensate for wear.

Before the new seals were installed, the rotor and seals were marked with a high speed grinder. This was done only to keep track of the dimensions of each seal and to ensure exact seal relocation in the rotor. Any logical method of marking is satisfactory for a test of this type and the method used will not be explained in detail. In correlating the wear data, the apex seals, side seals, and corner seals were represented by the numbers 1 through 3, 1 through 6, and 1 through 6 respectively.

Seal measurements were made with a precision micrometer at several different places on the seal. These measurements were then averaged to get a mean value of the dimension. The positions of measurement can best be seen by examination of Figure 12.

After initial seal measurements, the engine was reassembled and the wear test begun. The speed control weight was attached to the air vane arm of the throttle linkage and the precision throttle control was set to its fully open position, allowing the speed control to take over. The fuel line needle valve was removed and the purge button control screw on the regulator was released, allowing the regulator to act normally. Oil flow was started and set at 4 sec/drop and the engine was started as in the exhaust emissions tests. Load and speed settings were made by adjusting the dynamometer needle valve and by varying the number of BB-shot in the speed control weight. Speed was set to 3000 RPM and manifold vacuum was set at approximately 4 in. Hg. Due to results obtained in the exhaust emissions and efficiency tests, the air fuel ratio was adjusted by the coarse needle valve on the mixer to 18. At this operating point the engine was producing about 2.1 horsepower. The oil shutoff, high speed shutoff, and telephone signaling device were activated and the engine was allowed to run continuously. Engine hours were kept by the hourmeter wired through the vacuum operated microswitch.

Since the seal wear rates on propane operation and with the lubrication method used were unknown, the engine was stopped after 115 hours and the first seal measurement was made. No excessive wear was found and so the engine was reassembled with the same parts and run again under identical conditions. Three-hundred more hours were accumulated on the engine but severe carbon deposits were found on the faces of the rotor and covering about 1/6 the area of the spark plug port. The carbon

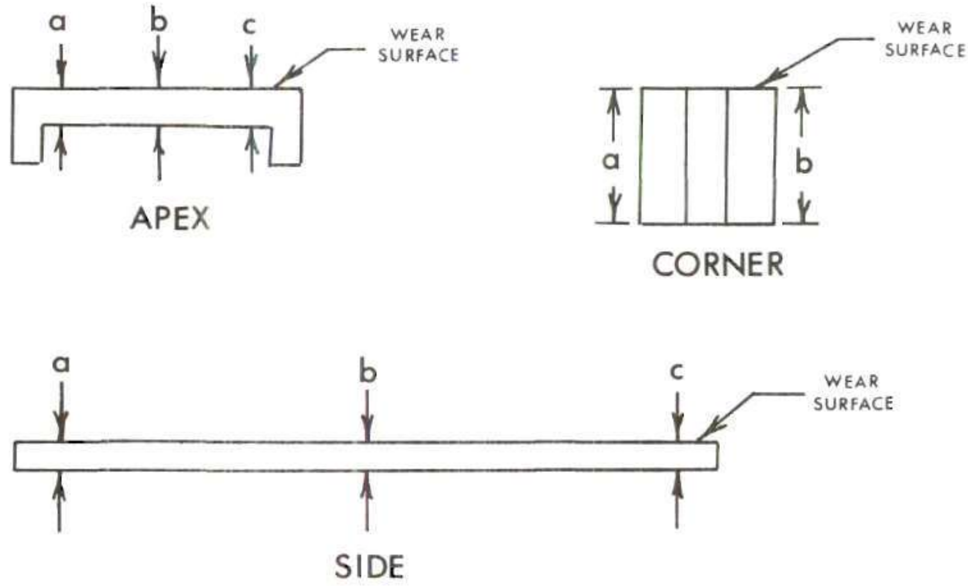

Figure 12. Measurement Positions of Rotor Gas Seals

was cleaned off and the engine reassembled and run at the same operating point, but with the oil rate reduced to 6 sec/drop in an effort to cut down on carbon buildup. The engine was then run for 308 more hours, at which point the wear study was concluded.

CHAPTER IV

RESULTS AND DISCUSSION

Power and Efficiency Data

Figure 13 shows the horsepower vs RPM curves produced by the gasoline powered engine at two load settings. As can be seen, horsepower climbs very nearly linearly with RPM at these two settings. This indicates almost constant engine torque throughout the entire RPM range. In these tests, carburetor jets were not altered from factory specifications and no attempt was made to measure, change, or hold constant, air-fuel ratios.

Figure 14 shows the thermal efficiency vs RPM curves for the gasoline powered engine at two load settings. Maximum efficiency is obtained at about 3500 RPM at 4 in. Hg., probably due to the inherent leaner operation at part load and mid-range RPM values.

Horsepower vs RPM curves for propane operation at WOT and 4 in. Hg. are shown in Figures 15 and 16. Again, for all mixture settings, power varies almost linearly with RPM. As is expected, horsepower output at a particular RPM increases slightly with richening of air-fuel mixture. If richening were continued past an air-fuel ratio of 13, a sharp decrease in power output would probably soon be noticed due to lack of air for complete combustion of the fuel.

Figures 17 and 18 show the thermal efficiency vs RPM curves for propane operation at 4 in. Hg. and WOT. In both sets of curves, maximum efficiency is obtained under the leaner mixture settings and efficiency

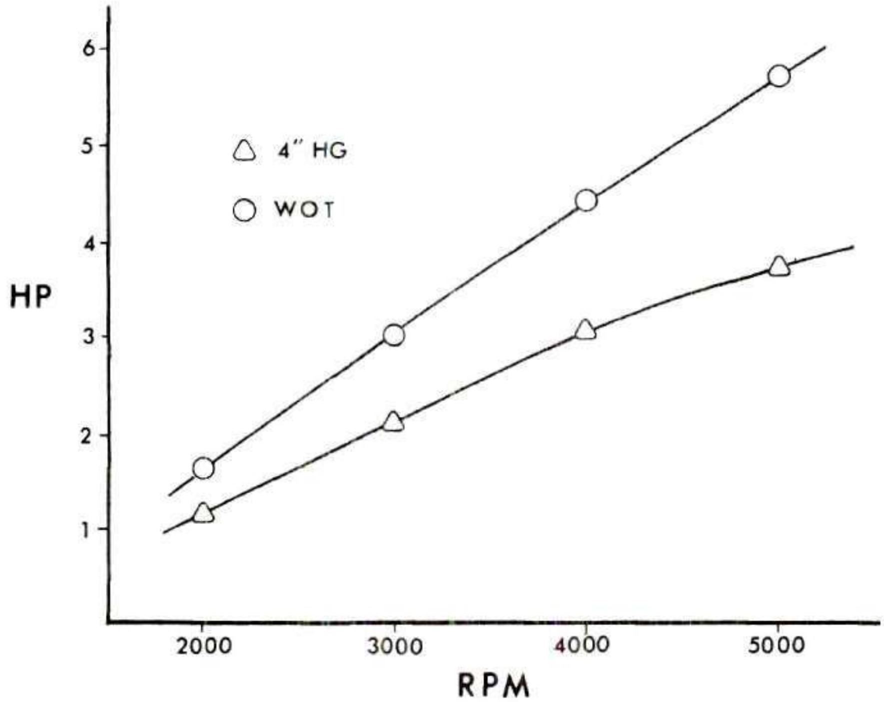

Figure 13. Horsepower vs RPM (Gasoline)

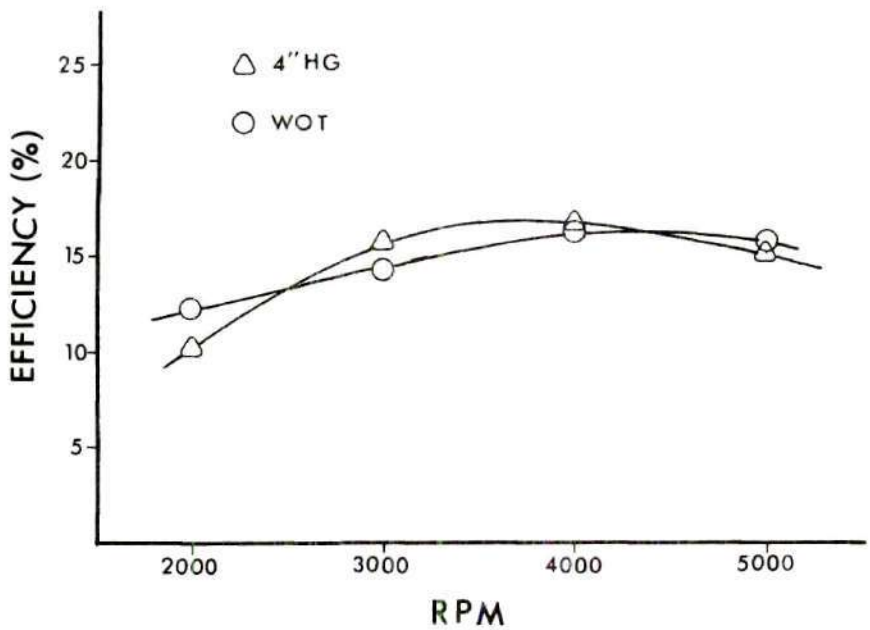

Figure 14. Efficiency vs RPM (Gasoline)

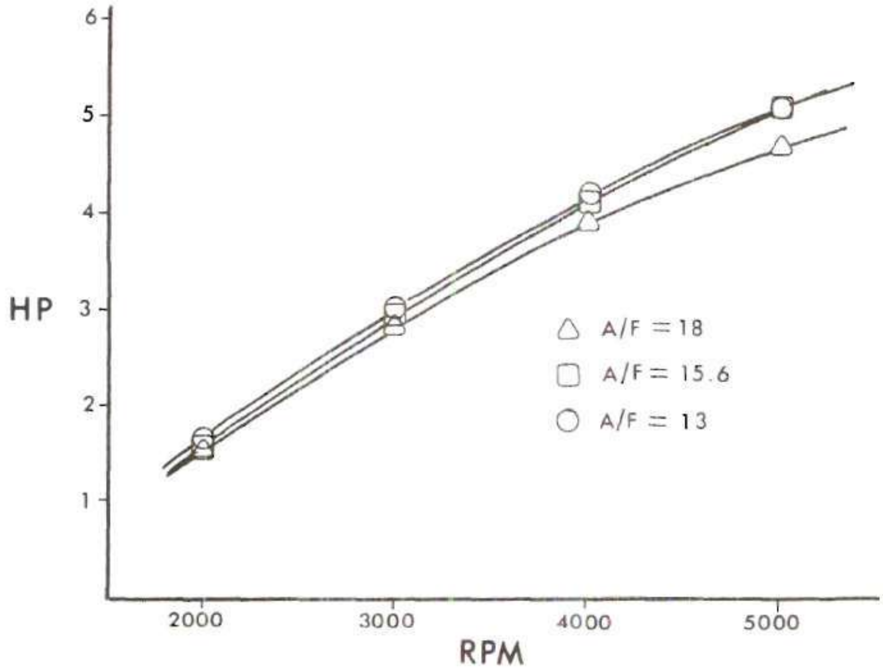

Figure 15. Horsepower vs RPM at WOT (Propane)

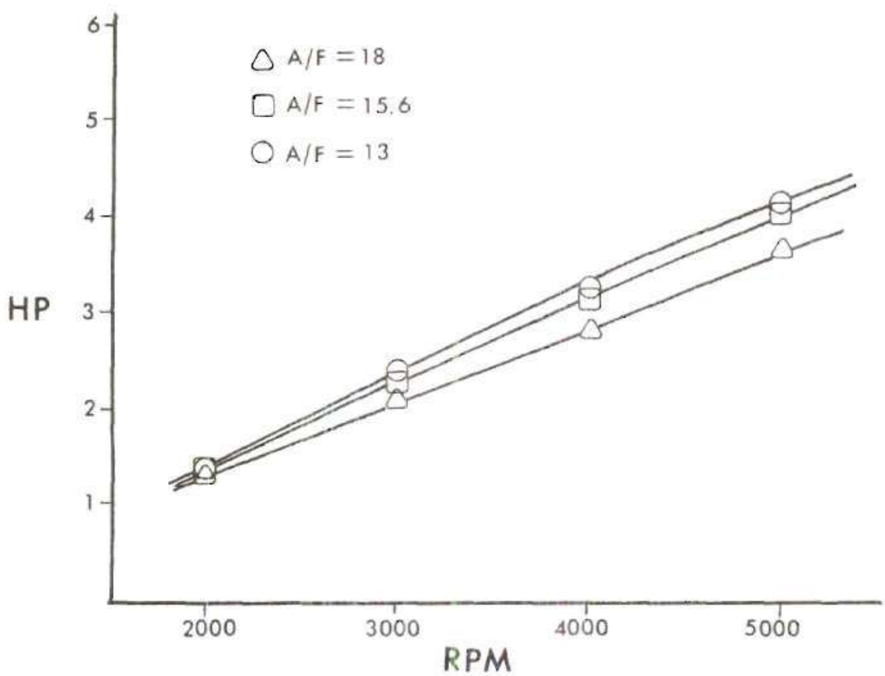

Figure 16. Horsepower vs RPM at 4 in. Hg. (Propane)

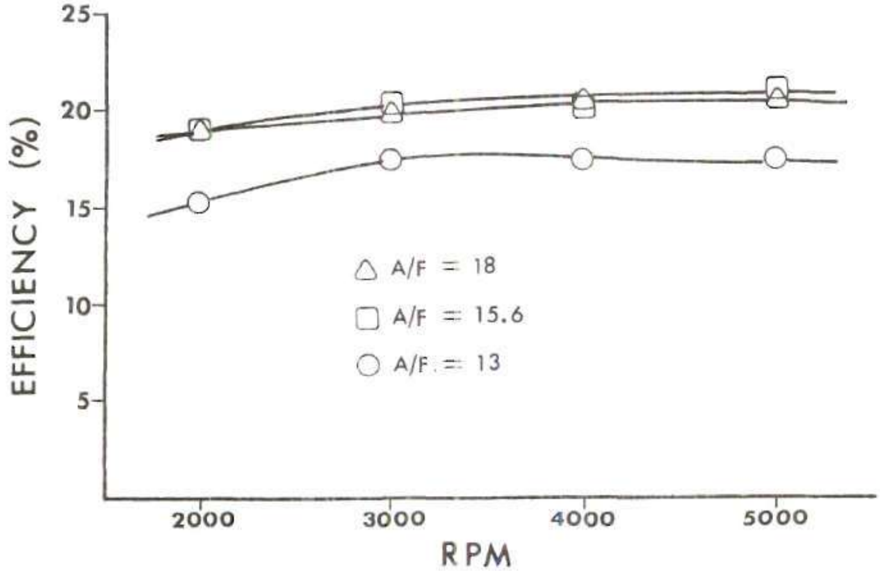

Figure 17. Efficiency vs RPM at 4 in. Hg. (Propane)

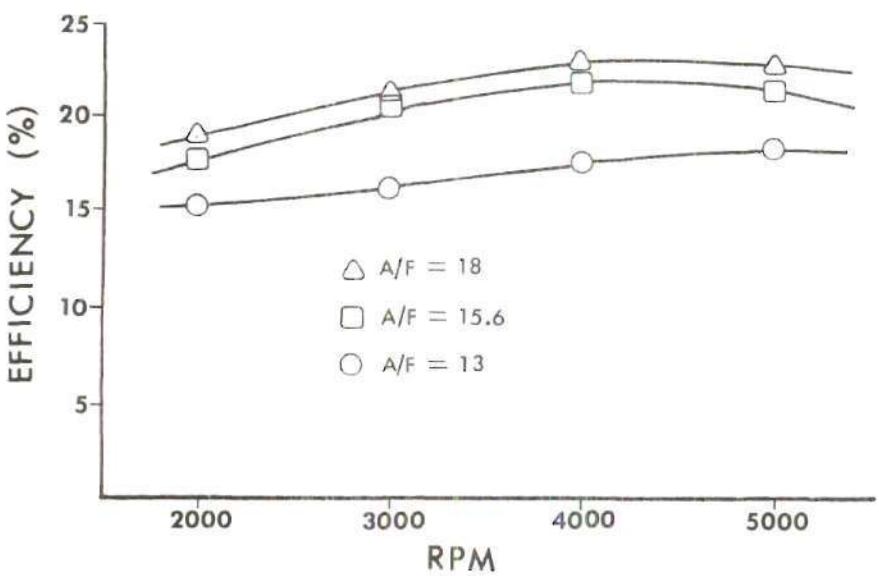

Figure 18. Efficiency vs RPM at WOT (Propane)

decreases rapidly as air-duel ratios are richened to 13, due to poor combustion associated with a deficiency of air. If leaning were continued past air-fuel ratios of 18, efficiencies would soon rapidly drop, due to lean misfire.

Exhaust Emissions

Exhaust emissions tests were conducted only on propane. Hydrocarbon, NO_x, CO, and CO_2, concentrations were monitored at three different loads, three different engine speeds, and three different air-fuel ratios, and a complete matrix of data points was determined. All data are presented in tabular form in Appendix G and the more meaningful data are plotted in Figures 19 through 24. All pollutant concentrations in the following graphs are expressed on a gm/HP-hr basis, and the method showing this conversion from dry volume concentrations obtained from the FID and NDIR analyzers is given in Appendix D.

Due to lack of time, no formal repeatability study was made on the exhaust emissions data. Spot checks for repeatability, however, were made and the results of these indicate an approximate repeatability of $\pm$ 15% in the following curves.

HC Emissions Variation with Air-Fuel Ratio

Figures 19 and 20 are typical of HC emissions variation with air-fuel mixture. In the part load curves it is shown that HC emissions decrease as air-fuel mixtures go from rich to stoichiometric, but increase as lean mixtures are approached. This increase is primarily due to lean misfire (audibly detected) and incomplete flame propagation in the combustion chamber. In the full load curves, however, HC concentrations

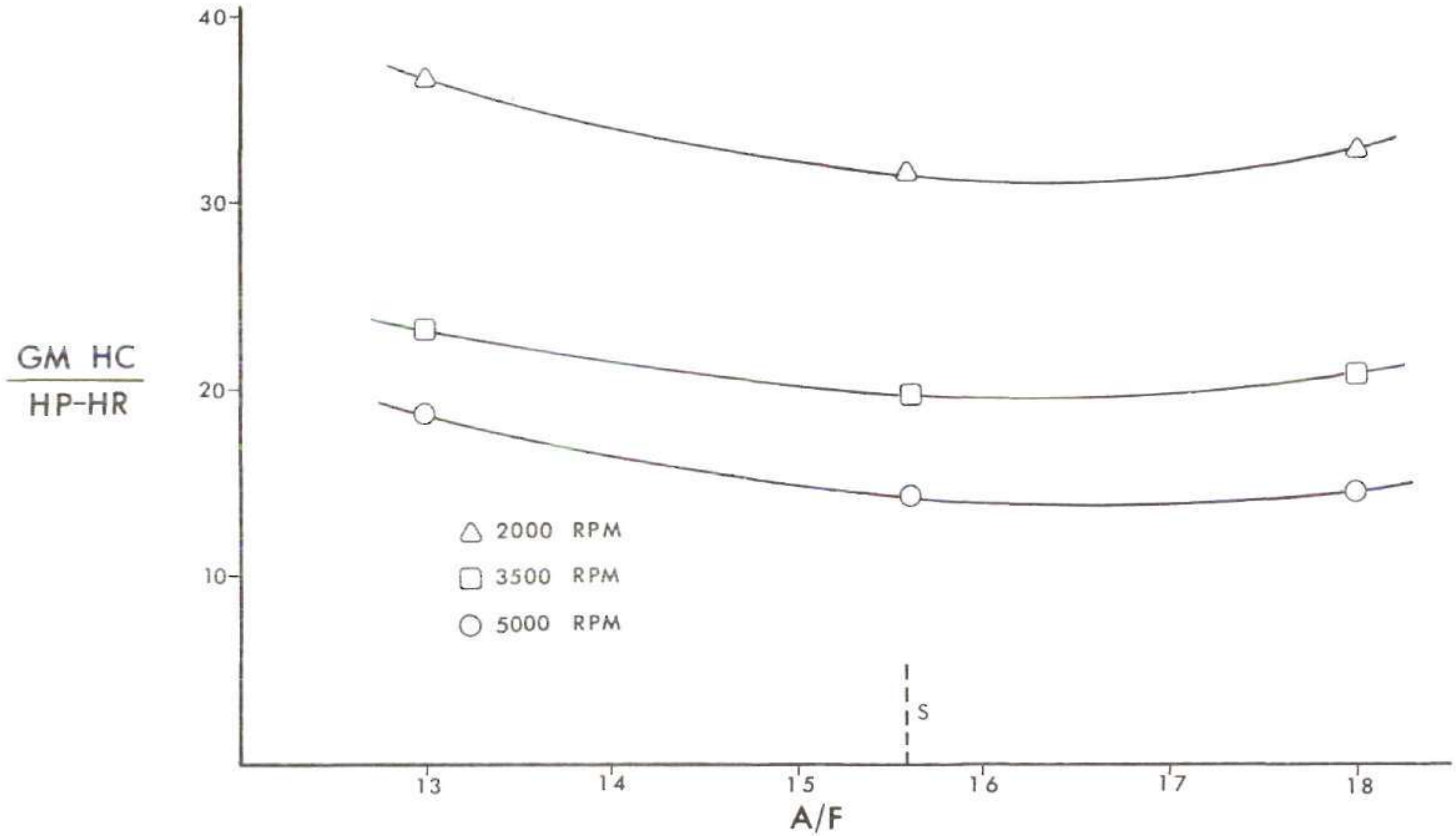

Figure 19. HC Emissions Variation with Air-Fuel Ratio (4 in. Hg.)

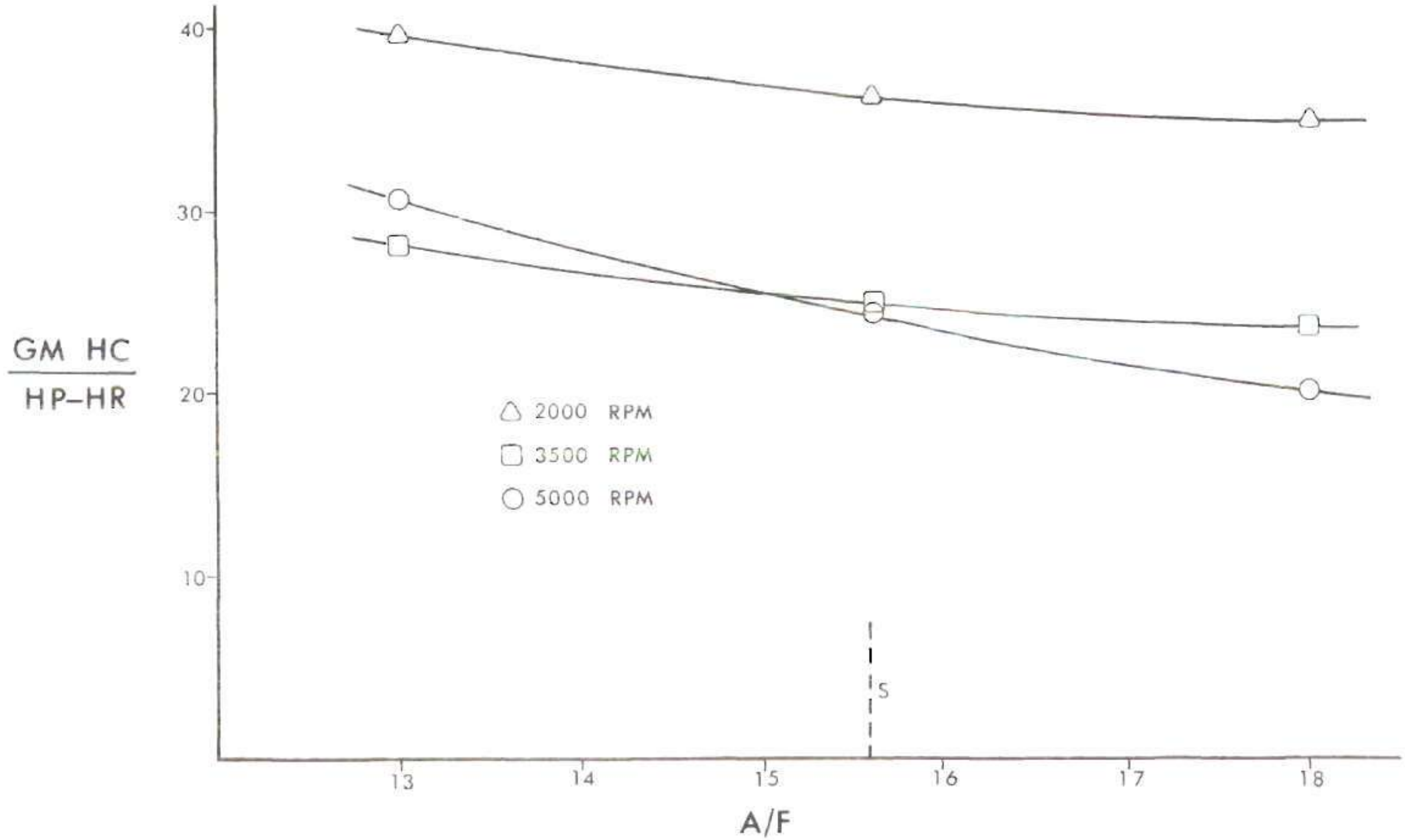

Figure 20. HC Emissions Variation with Air-Fuel Ratio (WOT)

continue to decrease as the mixture is leaned, primarily because of the inherent ability of the engine to run much more smoothly with less misfire at high load and high RPM. At both loads, the HC concentrations are higher at richer mixtures because the air necessary for complete combustion is not available.

Figures 19 and 20 also show the basic trends of HC emission reduction with increased RPM at constant load. This results mainly because increased engine speed causes increased turbulent mixing in the combustion chamber and allows less time for unburned air-fuel mixture to leak past the rotor gas seals and into the combustion chamber.

CO Emissions Variation with Air-Fuel Ratio

Figures 21 and 22 show variation of CO concentrations with air-fuel ratios at three different engine speeds. As can be seen, all the curves at both load settings have the same general shape and indicate the CO values are very high at rich air-fuel mixtures and drop off to relatively low values at lean mixtures. This is because at rich mixtures there is not enough air available to allow complete combustion to CO_2.

In comparing Figures 21 and 22, it is noticed that the load conditions and engine speeds at which the engine operates has very little effect on the average amounts of CO emitted. It is well known that CO emissions in I.C. engines have very little dependency on any parameter other than air-fuel ratio.

NO_x Emissions Variation with Air-Fuel Ratio

In engine exhaust emissions, NO_x concentrations reach their peak at the maximum combustion temperature and pressure. In I.C. engines,

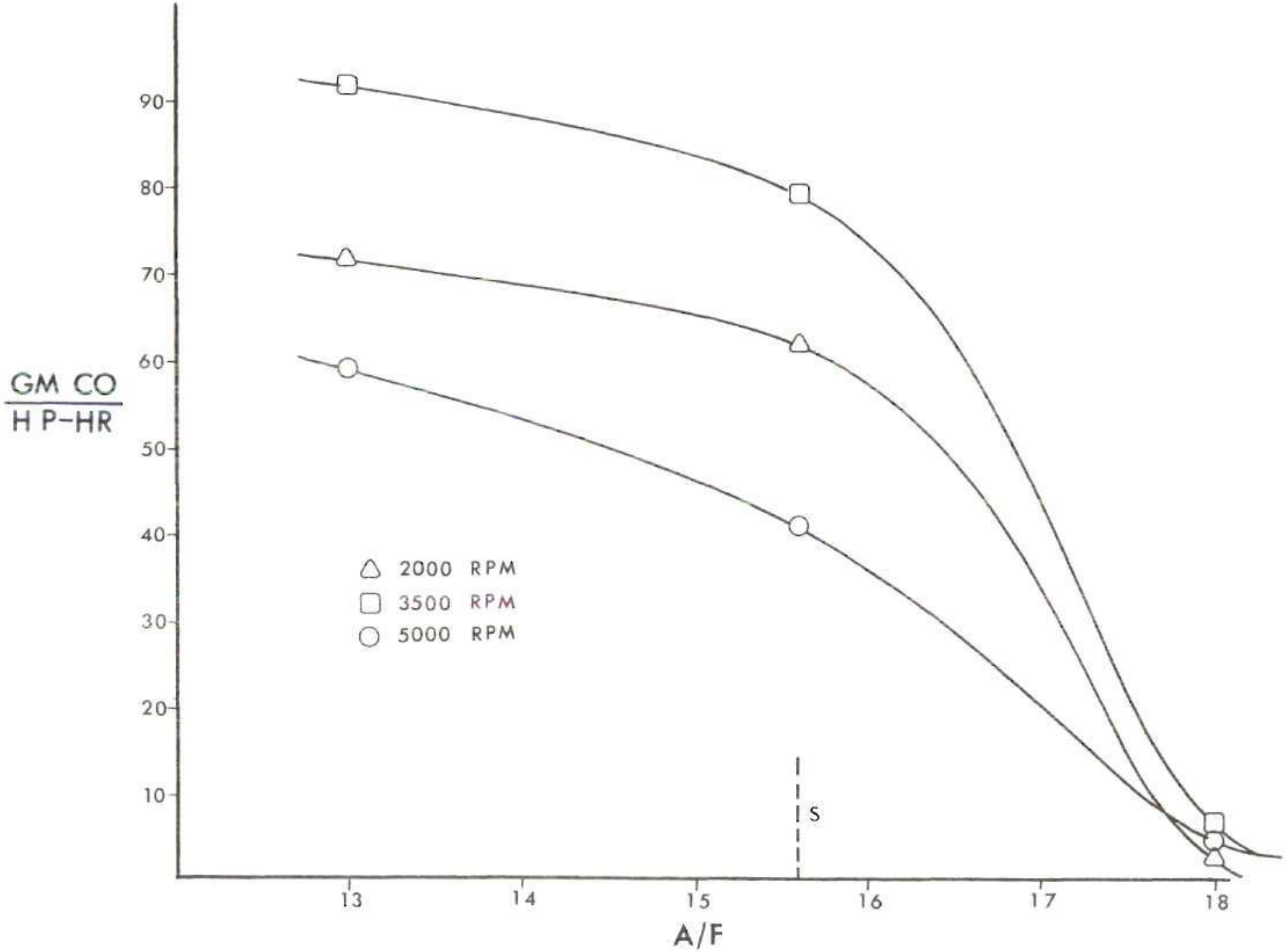

Figure 21. CO Emissions Variation with Air-Fuel Ratio (4 in. Hg.)

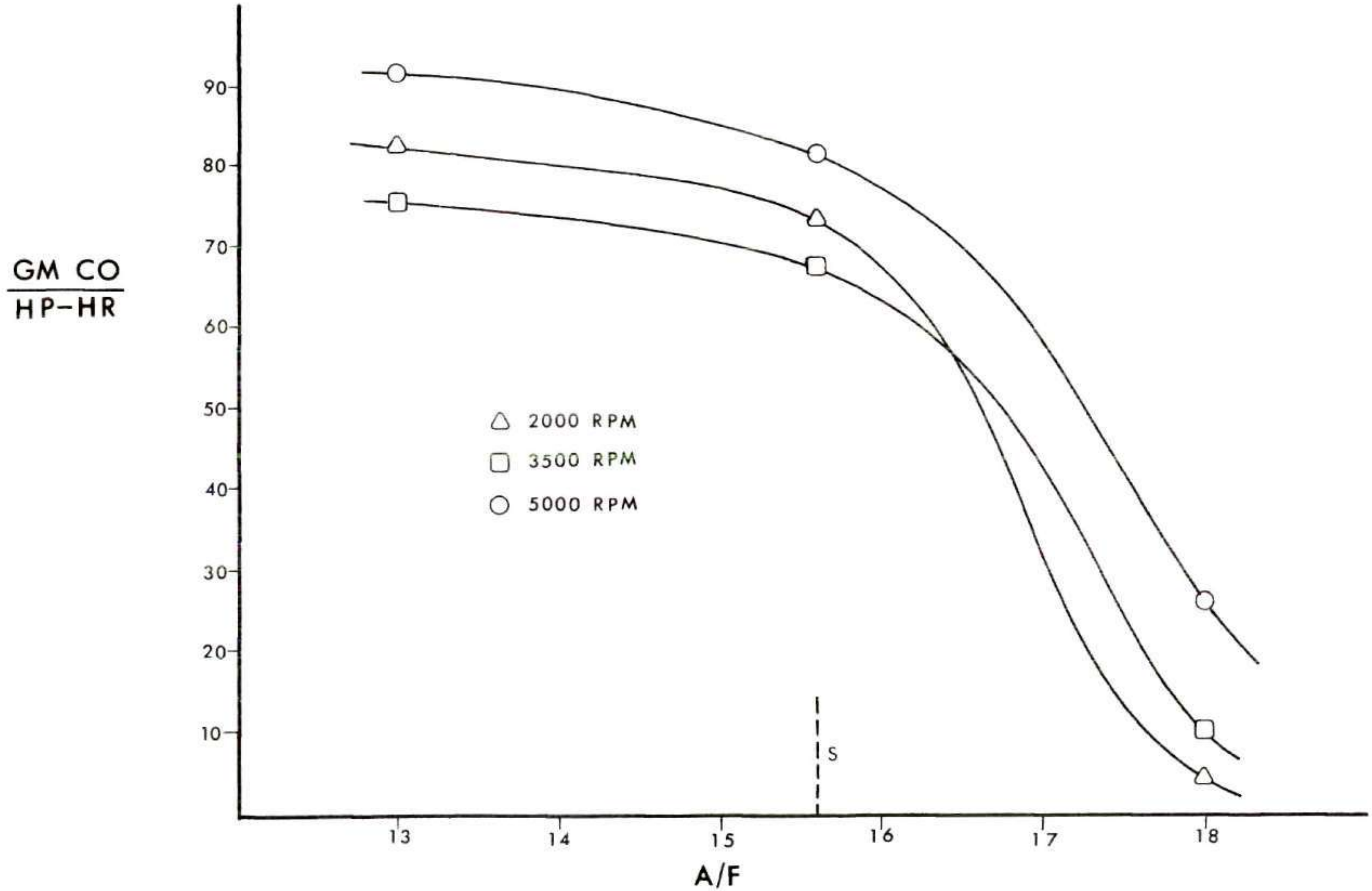

Figure 22. CO Emissions Variation with Air-Fuel Ratio (WOT)

these conditions are reached at approximately stoichiometric air-fuel ratios and, of course, under maximum load conditions.

Figure 23 shows NO_x emissions at WOT. As can be seen, in all cases the NO_x concentrations reach their peak values at stoichiometric air-fuel ratios and drop off to lower values at rich and lean conditions. At a particular air-fuel ratio, NO_x emissions are highest at the mid-range RPM values due to the higher mean effective pressures (MEP) developed.

Figure 24 shows NO_x exhaust emissions at partial load conditions. The decrease combustion temperatures and pressures obtained at 4 in. Hg. cause NO_x emissions to be extremely low. Values are so small that it is impossible to observe any peaking near stoichiometric air-fuel ratios.

Seal Wear

Figure 25 shows the average accumulated wear of the apex seals, corner seals, and side seals as a function of time. The data points were determined by averaging the wear at each of the measurement positions for all the seals of a particular type.

The general shape of the wear curves corresponds nicely to other Wankel engine seal wear curves found in the literature (11). The decrease in slope of the curves during the first several hundred hours of the test is due to rapid initial wear of the relatively rough, new seals. As the seals wear in, the wear rates decrease considerably.

Reference (6) gives the seal wear at which the factory recommends replacement which is .020 inches for the apex seals, .0118 inches for the corner seals, and .0079 inches for the side seals. If the wear curves are extropolated to these points, the time before seal replace-

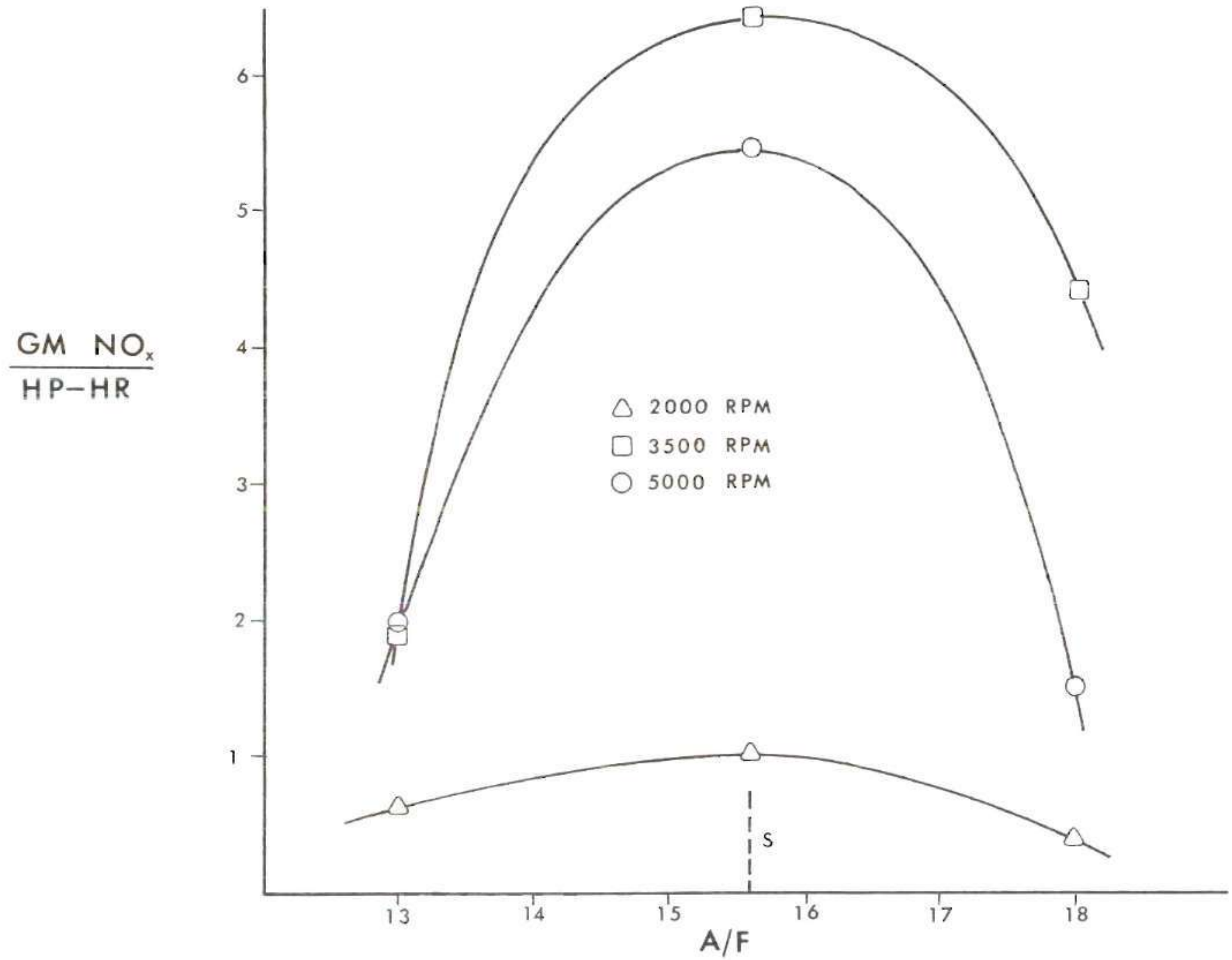

Figure 23. NO_x Emissions Variation with Air-Fuel Ratio (WOT)

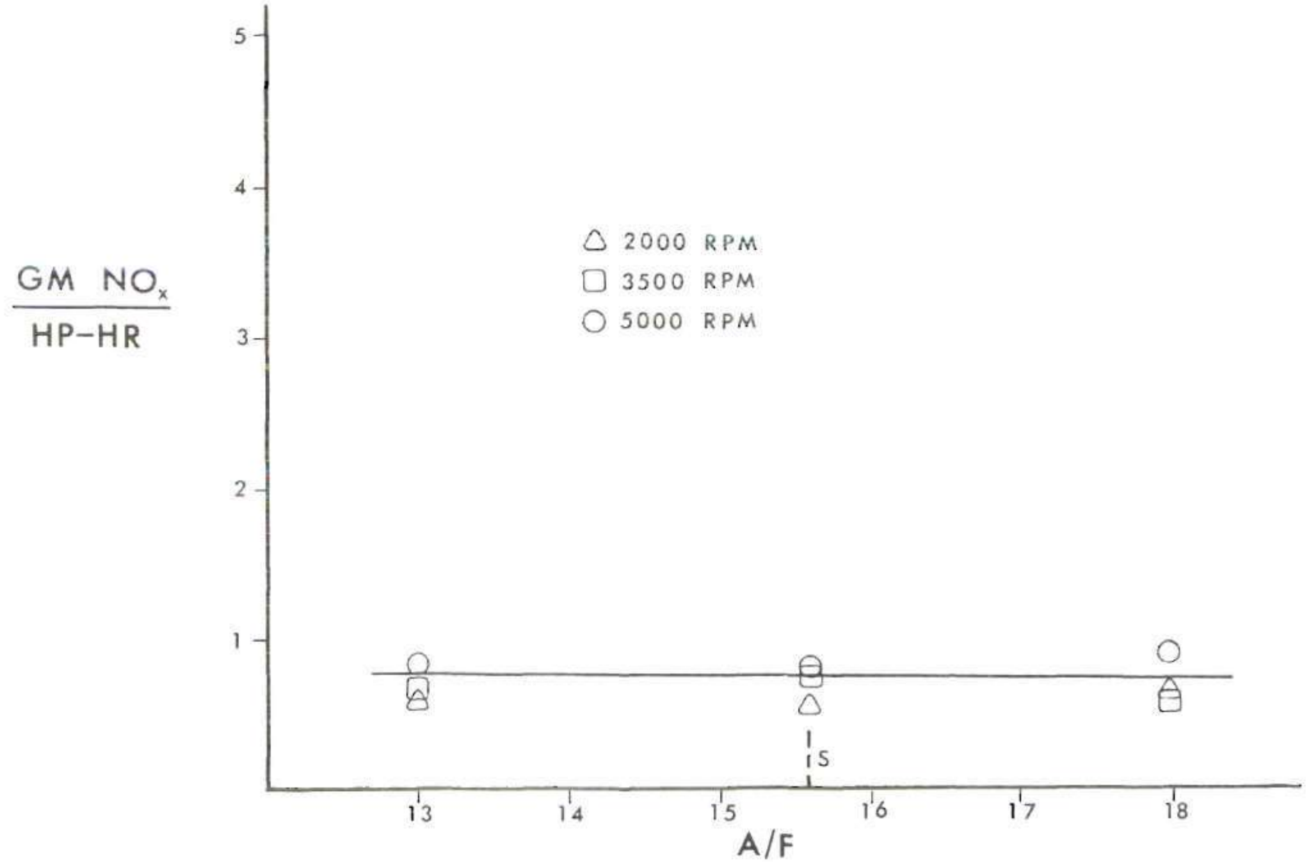

Figure 24. NO_x Emissions Variation with Air-Fuel Ratio (4 in. Hg.)

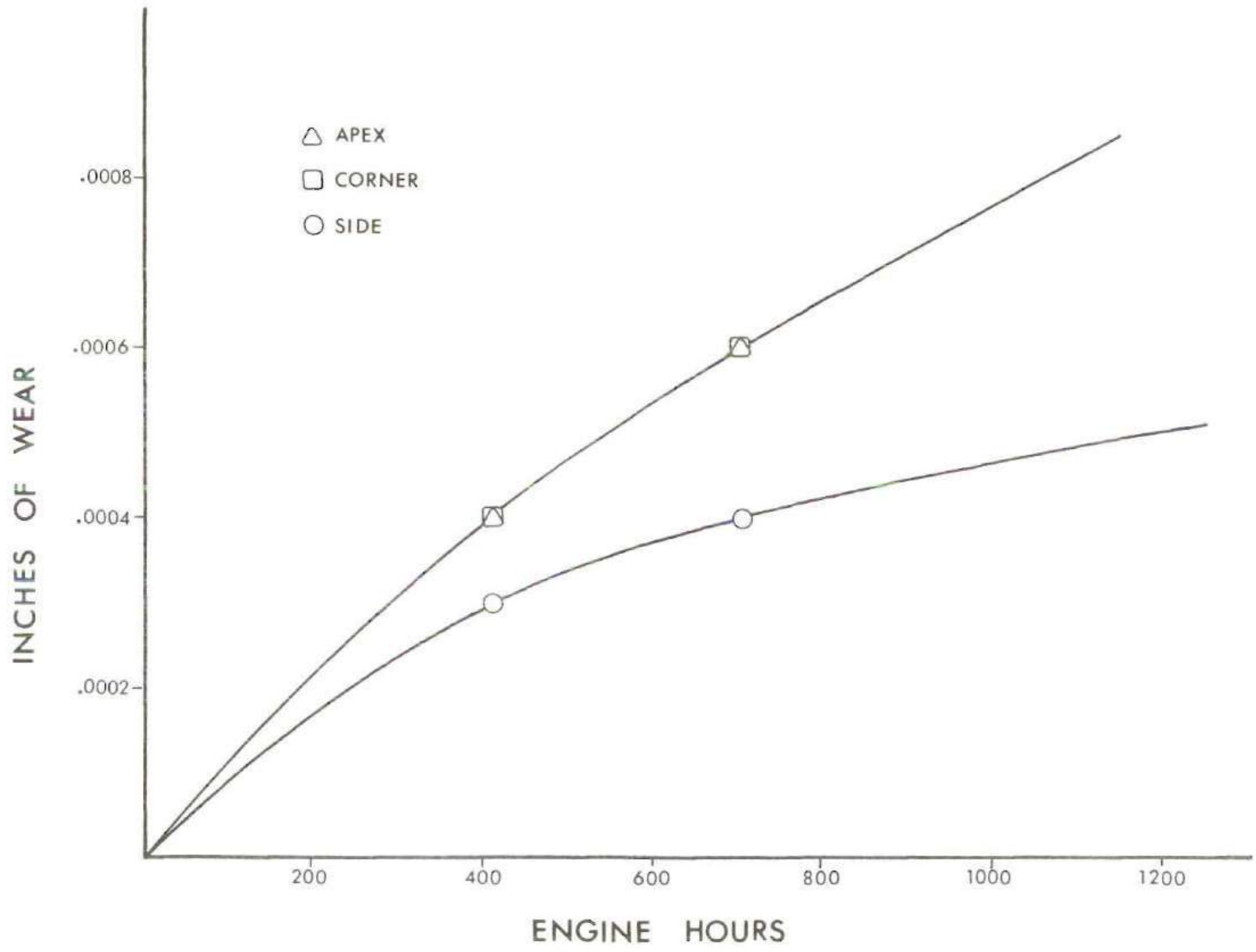

Figure 25. Seal Wear

ment is advised can be calculated. These times are 33,000 hours for apex seals, 19,000 hours for corner seals, and 53,000 hours for side seals.

It should be noted that the engine was not new when the test was begun and that some scoring and chattermarks were present on the side housings and epitrochoidal bore respectively at the beginning of the tests on gaseous fuel. Had a new engine been used for the test, hours of operation longer than the ones stated above could have possibly been extropolated.

Emissions Comparisons of Various Heating Systems

Since the type of engine studied is being considered as a prime mover to drive a gaseous powered heat pump, it is necessary to compare exhaust emissions from this type of system with those of conventional heating systems. The systems to be compared with a gaseous heat pump having a COP of 1.6 are electric resistance heating with a COP of .3, electric heat pump heating with a COP of .6, and gas furnace heating with a COP of .8 (4). It is assumed that electricity for the electric systems is supplied by a coal fired power plant burning 3 percent sulphur, bituminous coal, the type typically used in Georgia.

Table 2(a) shows the exhaust emissions from the different systems mentioned. The comparison is based on the mass of pollutant emitted per 10^5 BTU of useful heat produced. Hydrocarbon, NO_x, CO, and SO_x data for the electric and gas furnace systems were obtained from reference (12) and were given in (lbm pollutant)/(ton coal input) and (lbm pollutant)/ (ft^3 gas input) respectively. Particulate data were based on the maxi-

Table 2(a). Emissions of Various Heating Systems (lbm pollutant)/(10^5 BTU useful)

Method of Heating	Particulates	NO_x	HC	CO	SO_x	Total
Resistance	.06	.16	.0023	.0088	1.0	1.231
Electric Heat Pump	.03	.08	.0012	.0044	.5	.615
Gas Furnace	0	.0095	.00095	.0021	0	.0125
Gaseous Heat Pump	0	.0066	.23	.069	0	.305

Table 2(b). Toxicity Weighted Emissions of Various Heating Systems (lbm pollutant)/(10^5 BTU useful)

Method of Heating	Particulates	NO_x	HC	CO	SO_x	Total
Toxicity Weighting Factor	1.06	.8	.5	.008	1.0	
Resistance	.064	.128	.0012	.000071	1.0	1.193
Electric Heat Pump	.032	.064	.0006	.000035	.5	.597
Gas Furnace	0	.0076	.00048	.000017	0	.0081
Gaseous Heat Pump	0	.0053	.12	.000055	0	.126

mum controlled emission rate in Georgia of (.18 lbm particulates)/ (10^6 BTU input). Knowing the heating values of natural gas and coal, the emissions were converted to (lbm pollutant)/(10^5 BTU useful heat produced) by the method shown in Appendix F. Hydrocarbon data was given on a methane basis, but was converted to a hexane basis for proper comparison.

The emissions data for the gaseous heat pump system were obtained from the engine emissions data determined in this work and were also converted to (lbm pollutant)/(10^5 BTU usable heat produced). Appendix F shows this conversion also.

As can be seen, the HC and CO emissions of the gaseous powered heat pump system are much higher than with any of the other systems. These emissions, however, could possibly be substantially reduced with the use of a catalytic muffler on the engine.

Emissions of NO_x and SO_x, however, are considerably less than those of the two electric systems. Since the toxicity (13) of SO_x is 125 times that of CO, two times that of HC, and 1.25 times that of NO_x, it appears that the electric systems create much more of a health problem than the gaseous heat pump system, due to the high SO_x levels of the electric systems.

Table 2(b) shows the total toxicity weighted emissions from each system. The gaseous fueled heat pump emits five to ten times less toxic material into the atmosphere than electric systems. As is expected, however, all systems are much higher pollutors than gas furnace heating.

CHAPTER V

CONCLUSIONS

Based on a power, efficiency, and exhaust emissions standpoint, the best engine operating conditions are found to be when lean air-fuel mixtures are used. It is seen that at an air-fuel ratio of 18, NO_x and CO concentrations reach relatively low levels. Under part load conditions, HC concentrations are slightly higher with lean air-fuel ratios, but not enough so as to warrent operation at richer mixtures. Hydrocarbon emissions at full load reach their lowest levels under lean conditions. An important point to notice is that maximum engine thermal efficiency is obtained under lean conditions and the power decrease at a particular engine speed is very small. In general, it can be said that a lean air-fuel mixture gives best economy and cleanest exhaust without great sacrifices in engine performance.

The seal wear curves give strong indication that a gaseous powered Wankel engine of the type tested can be used as a reliable, stationary prime mover. Judging from the extropolated wear curve of the corner seals, it appears that approximately 19,000 hours of operation can be obtained before seal replacement. If the engine was used in a gaseous heat pump of the kind mentioned, assuming 3000 hours of operation per year, the system could be expected to perform satisfactorily for over six years without major maintenance from the owner. Judging from the apex seals, 33,000 hours of operation can be obtained. These long seal lifetimes appear to make the system quite desirable, and competitive with electric systems in reliability.

APPENDIX A

SAMPLE CALCULATIONS OF HORSEPOWER AND TORQUE VALUES

Horsepower and torque values are calculated directly from dynamometer scale and tachometer readings:

$$F = \text{scale reading} = 8.0 \text{ lbf}$$

$$N = \text{tachometer reading} = 3500 \text{ RPM}$$

$$HP = \frac{(F)(N)}{8266}$$

$$HP = \frac{(8.0)(3500)}{8266} = 3.39$$

$$\text{Torque} = (.635)(F)$$

$$\text{Torque} = (.635)(8.0) = 5.1 \text{ ft lbf}$$

APPENDIX B

SAMPLE CALCULATION OF EFFICIENCY OF THE GASOLINE FUELED ENGINE

HP = 3.8

Q_g = gasoline volume flow rate = .545 ml/sec

Assuming the density of gasoline is 1.6 lbm/lit and gasoline has a heating value of 18,800 BTU/lbm, the thermal efficiency, η, is found by:

$$\eta = \frac{(\mathrm{HP})(33{,}000\ \mathrm{ft\ lbf/HP\ min})(1000\ \mathrm{ml/lit})(100\%)}{(Q_g\ \mathrm{ml/sec})(60\ \mathrm{sec/min})(1.6\ \mathrm{lbm/lit})(18{,}800\ \mathrm{BTU/lbm})(778\ \mathrm{ft\ lbf/BTU})}$$

$$= \frac{(2.35)(\mathrm{HP})}{(Q_g)}$$

$$\eta = \frac{(2.35)(3.8)}{(.545)} = 16.4\%$$

APPENDIX C

SAMPLE CALCULATION OF EFFICIENCY OF THE PROPANE FUELED ENGINE

Q_p = fuel volume flow rate = .50 ft^3/min

T = fuel temperature = 85 F = 545 R

P = fuel pressure = 15.8 lbf/in^2

HP = 5.14

Molecular weight of propane = 44 lbm/lbmole

Heating value of propane = 19,800 BTU/lbm

Step 1:

The fuel mass flow rate, $\dot{m}_p$, is calculated from the ideal gas equation:

$$\dot{m}_p = \frac{(P\ lbf/in^2)(Q_p\ ft^3/min)(44\ lbm/lbmole)(60\ min/hr)(144\ in^2/ft^2)}{(T^oR)(1545\ ft\ lbf/lbmole^oR)}$$

$$= \frac{(246.6)(P)(Q_p)}{T}$$

$$\dot{m}_p = \frac{(246.6)(15.8)(.50)}{545} = 3.57\ lbm/hr$$

Step 2:

Knowing the fuel mass flow rate, the thermal efficiency, η, is calculated:

$$\eta = \frac{(HP)(33{,}000\ ft\ lbf/HP\ min)(60\ min/hr)(100\%)}{(\dot{m}_p\ lbm/hr)(19{,}800\ BTU/lbm)(778\ ft\ lbf/BTU)}$$

$$= \frac{(12.85)(HP)}{m_P}$$

$$\eta = \frac{(12.85)(5.14)}{3.57} = 18.6\%$$

APPENDIX D

SAMPLE CALCULATIONS OF MASS EMISSIONS FROM DRY VOLUME CONCENTRATIONS

HP = 3.39

A/F = air-fuel ratio = 18

m_p = fuel mass flow rate = 1.98 lbm/hr

[HC] = dry volume percent HC on a C_6H_{14} basis = .1867

[CO] = dry volume percent CO = .25

$[CO_2]$ = dry volume percent CO_2 = 11.3

[NO] = dry volume percent NO = .0650

$[HC]_w$ = wet volume percent HC on a C_6H_{14} basis

$[CO]_w$ = wet volume percent CO on a wet basis

$[CO_2]_w$ = wet volume percent CO_2 on a wet basis

$[NO]_w$ = wet volume percent NO on a wet basis

Step 1:

Using the equations for propane found in reference 10, the correction factor, C_f, to convert dry concentrations to a wet basis is calculated:

$$C_f = \frac{[HC]_w}{[HC]} \quad \text{where}$$

$$[HC]_w = \frac{[HC]}{a - [HC]\, b},$$

$$a = 1 + \frac{\frac{1.335}{N}}{\frac{[CO]}{3.8[CO_2]} + 1},$$

$$b = \frac{.07}{\frac{[CO]}{3.8[CO_2]} + 1}, \text{ and}$$

$$N = \frac{100}{6\,[HC] + [CO] + [CO_2]}$$

Substituting in appropriate dry volume concentrations gives:

$$a = 1.168$$

$$b = .070$$

$$[HC]_w = .1607$$

$$C_f = \frac{[HC]_w}{[HC]} = \frac{.1607}{.1867} = .86$$

Step 2:

All dry concentrations are multiplied by C_f to obtain volume concentrations on a wet basis.

$$[HC]_w = (.1867)\ (.86) = .1607$$

$$[CO]_w = (.25)\ (.86) = .22$$

$$[CO_2]_w = (11.3)\ (.86) = 9.7$$

$$[NO]_w = (.0650)\ (.86) = .0559$$

Step 3:

Mass emissions on a gm/HP-hr basis are now calculated:

$$\frac{\text{gm emission}}{\text{HP-hr}} = \frac{(454)\ \text{gm/lbm})(\dot{m}_p\ \text{lbm/hr})(1 + A/F)[\text{emission}]_w(M_e)}{(HP)(M_{mix})}$$

where M_e = molecular weight of particular emission (lbm/lbmole)

M_{mix} = molecular weight of the air-fuel mixture

$$= \frac{(1276)(1 + A/F)}{(44)(A/F) + 29} \quad \text{(lbm/lbmole)}$$

Substituting in appropriate values gives:

$$\frac{\text{gm NO}}{\text{HP-hr}} = 2.86$$

$$\frac{\text{gm CO}}{\text{HP-hr}} = 10.2$$

$$\frac{\text{gm } CO_2}{\text{HP-hr}} = 729$$

$$\frac{\text{gm HC}}{\text{HP-hr}} = 23.6$$

APPENDIX E

DERIVATION OF EQUATION FOR CALCULATING AIR-FUEL RATIOS FROM INTAKE MIXTURE ANALYSIS

B = volume fraction HC on a per carbon basis

P = pressure of intake mixture

T = temperature of intake mixture

M_{air} = molecular weight of air = 29 lbm/lbmole

M_p = molecular weight of propane = 44 lbm/lbmole

V = total volume of air-fuel mixture

A/F = air-fuel ratio = $\frac{\text{lbm air}}{\text{lbm propane}}$

ρ_{air} = density of air

ρ_p = density of propane

From the ideal gas equation, the densities of air and propane are expressed as:

$$\rho_{air} = \frac{(P)(M_{air})}{(R)(T)}$$

$$\rho_p = \frac{(P)(M_p)}{(R)(T)}$$

Since there are three carbon atoms in one propane molecule then:

$B/3$ = volume fraction of propane in the mixture

$1-B/3$ = volume fraction of air in the mixture

The masses of the propane and air are calculated by:

lbm air $= (V)(1-B/3)(\rho_{air})$

lbm propane $= (V)(B/3)(\rho_p)$

An expression for the air-fuel ratio as a function of volume fraction HC on a per carbon basis is obtained by combing expressions:

$$A/F = \frac{\text{lbm air}}{\text{lbm propane}} = \frac{\dfrac{(V)(1-B/3)(P)(M_{air})}{(R)(T)}}{\dfrac{(V)(B/3)(P)(M_p)}{(R)(T)}}$$

$$= \frac{(1-B/3)(M_{air})}{(B/3)(M_p)}$$

$$= 1.97/B - .656$$

APPENDIX F

SAMPLE CALCULATION FOR COMPARING EMISSIONS FROM DIFFERENT HEATING SYSTEMS

For electric heating:

lbm NO_x/ton imput = 18

Heating value of coal = 19000 BTU/lbm

COP of resistance heating = .3 BTU useful/BTU input

$$\frac{\text{lbm } NO_x}{10^5 \text{ BTU useful}} = \frac{(\text{lbm } NO_x/\text{ton input})(10^5)}{(19000 \text{ BTU/lbm})(2000 \text{ lbm/ton})(\text{COP BTU useful/BTU input})}$$

$$= \frac{(18)(10^5)}{(19000)(2000)(.3)} = .16$$

For the gaseous heat pump:

HP = 5.14

$\dot{m}_p$ = fuel mass consumption rate = 3.57 lbm/hr

$$\frac{\text{gm } NO_x}{\text{HP-hr}} = 4.39$$

Heating value of propane = 19,800 BTU/lbm

COP of gaseous heat pump = 1.6 BTU useful/BTU input

$$\frac{\text{lb } NO_x}{10^5 \text{ BTU input}} \quad \frac{(\text{gm } NO_x/\text{HP-hr})(\text{HP})(10^5)}{(454 \text{ gm/lbm})(m_p \text{ lbm/hr})(19800 \text{ BTU/lbm})(\text{COP BTU useful/BTU input})}$$

$$= \frac{(4.39)(5.14)(10^5)}{(454)(3.57)(19800)(1.6)} = .0044$$

APPENDIX G

TABLES

Table G1. Horsepower and Efficiency (Gasoline)

RPM	Load	Horsepower	Efficiency (%)
2000	WOT	1.63	12.1
3000	WOT	3.03	14.4
4000	WOT	4.45	16.3
5000	WOT	5.75	15.7
2000	4" Hg	1.14	10.2
3000	4" Hg	2.12	15.7
4000	4" Hg	3.10	16.7
5000	4" Hg	3.78	15.4

Table G2. Horsepower and Efficiency (Propane)

RPM	Load	A/F	Horsepower	Efficiency (%)
2000	WOT	13	1.60	15.3
3000	WOT	13	2.98	16.2
4000	WOT	13	4.21	17.5
5000	WOT	13	5.14	18.2
2000	WOT	15.6	1.58	17.7
3000	WOT	15.6	2.94	20.5
4000	WOT	15.6	4.19	21.8
5000	WOT	15.6	5.14	21.2
2000	WOT	18	1.54	19.0
3000	WOT	18	2.81	21.1
4000	WOT	18	3.92	22.8
5000	WOT	18	4.72	22.5
2000	4" Hg	13	1.27	15.3
3000	4" Hg	13	2.38	17.5
4000	4" Hg	13	3.27	17.5
5000	4" Hg	13	4.17	17.7
2000	4" Hg	15.6	1.27	18.8
3000	4" Hg	15.6	2.29	20.3
4000	4" Hg	15.6	3.17	20.4
5000	4" Hg	15.6	4.05	21.6
2000	4" Hg	18	1.26	18.8
3000	4" Hg	18	2.07	20.0
4000	4" Hg	18	2.81	20.5
5000	4" Hg	18	3.69	20.8

Table G3. Exhaust Emissions (WOT and 4" Hg)

RPM	Load	A/F	C_f	$\frac{\text{gm HC}}{\text{HP-hr}}$	$\frac{\text{gm } CO_2}{\text{HP-hr}}$	$\frac{\text{gm CO}}{\text{HP-hr}}$	$\frac{\text{gm } NO_x}{\text{HP-hr}}$
2000	WOT	13	.88	39.7	575	82.5	.64
3500	WOT	13	.88	28.1	572	75.8	1.90
5000	WOT	13	.88	30.6	505	91.4	1.97
2000	WOT	15.6	.86	36.4	792	73.5	1.03
3500	WOT	15.6	.85	25.0	680	67.7	6.45
5000	WOT	15.6	.84	24.6	690	81.7	5.50
2000	WOT	18	.86	35.0	925	4.7	.42
3500	WOT	18	.86	23.8	736	10.3	4.43
5000	WOT	18	.86	20.2	746	26.3	1.52
2000	4" Hg	13	.89	36.6	572	71.5	.58
3500	4" Hg	13	.88	23.1	528	91.7	.66
5000	4" Hg	13	.87	18.7	559	58.9	.84
2000	4" Hg	15.6	.85	31.6	757	61.7	.55
3500	4" Hg	15.6	.85	19.7	749	79.0	.79
5000	4" Hg	15.6	.84	14.3	727	40.8	.80
2000	4" Hg	18	.87	32.9	831	2.4	.63
3500	4" Hg	18	.87	20.9	815	6.4	.60
5000	4" Hg	18	.87	14.5	794	4.4	.91

Table G4. No-load Exhaust Emissions (Dry)

RPM	Load	A/F	HC(ppm)	CO_2(%)	CO(%)	NO_x(ppm)
2000	No	13	12000	5.0	4.40	138
3500	No	13	11200	5.4	4.45	153
5000	No	13	9600	6.0	3.90	107
2000	No	15.6	9067	7.0	1.47	169
3500	No	15.6	8800	7.3	1.37	169
5000	No	15.6	8533	7.5	1.6	123
2000	No	18	8000	6.4	.15	153
3500	No	18	7733	7.2	.15	138
5000	No	18	7733	7.2	.25	107

Table G5. Apex Seal Wear Data

Seal Number	Hours of Operation	Dimensions (inches) a	b	c	Wear (inches) a	b	c
1	0	.2674	.2677	.2683	0	0	0
1	415	.2672	.2676	.2680	.0002	.0001	.0003
1	723	.2670	.2675	.2677	.0004	.0002	.0006
2	0	.2680	.2677	.2673	0	0	0
2	415	.2676	.2673	.2669	.0004	.0004	.0004
2	723	.2674	.2671	.2667	.0006	.0006	.0006
3	0	.2706	.2707	.2710	0	0	0
3	415	.2703	.2703	.2707	.0003	.0004	.0003
3	723	.2700	.2701	.2704	.0006	.0006	.0006

Table G6. Corner Seal Wear Data

Seal Number	Hours of Operation	Dimension (inches) a	b	Wear (inches) a	b
1	0	.2175	.2174	0	0
1	415	.2172	.2169	.0003	.0005
1	723	.2171	.2167	.0004	.0007
2	0	.2190	.2188	0	0
2	415	.2187	.2184	.0003	.0004
2	723	.2185	.2182	.0005	.0006
3	0	.2188	.2191	0	0
3	415	.2186	.2187	.0002	.0004
3	723	.2183	.2184	.0005	.0007
4	0	.2177	.2179	0	0
4	415	.2172	.2176	.0005	.0003
4	723	.2171	.2174	.0006	.0005
5	0	.2169	.2172	0	0
5	415	.2166	.2170	.0003	.0002
5	723	.2163	.2167	.0006	.0005
6	0	.2177	.2173	0	0
6	415	.2170	.2168	.0007	.0005
6	723	.2167	.2165	.0010	.0008

Table G7. Side Seal Wear Data

Seal Number	Hours of Operation	Dimensions (inches) a	b	c	Wear (inches) a	b	c
1	0	.0922	.0924	.0923	0	0	0
1	415	.0920	.0921	.0920	.0002	.0003	.0003
1	723	.0918	.0918	.0917	.0004	.0006	.0006
2	0	.0926	.0925	.0925	0	0	0
2	415	.0925	.0922	.0922	.0001	.0003	.0003
2	723	.0924	.0922	.0922	.0002	.0003	.0003
3	0	.0928	.0929	.0929	0	0	0
3	415	.0926	.0926	.0927	.0002	.0003	.0002
3	723	.0926	.0925	.0926	.0002	.0004	.0003
4	0	.0927	.0927	.0927	0	0	0
4	415	.0925	.0924	.0925	.0002	.0003	.0002
4	723	.0924	.0922	.0923	.0003	.0005	.0004
5	0	.0928	.0927	.0926	0	0	0
5	415	.0926	.0923	.0922	.0002	.0002	.0004
5	723	.0924	.0922	.0921	.0004	.0005	.0005
6	0	.0924	.0924	.0924	0	0	0
6	415	.0921	.0919	.0918	.0003	.0005	.0006
6	723	.0918	.0918	.0917	.0006	.0006	.0007

Table G8. Overall Average Seal Wear

Seal	Hours of Operation	Dimensions (inches)	Wear (inches)
Apex	0	.2688	0
Apex	415	.2684	.0004
Apex	723	.2682	.0006
Corner	0	.2810	0
Corner	415	.2176	.0004
Corner	723	.2174	.0006
Side	0	.0926	0
Side	415	.0923	.0003
Side	723	.0922	.0004

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